Vol. X. — No. 5. May 1928.

BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS

ASSOCIATION

(ENGLISH EDITION)

[625 .4 (01]

The railway formula for extra-light ultra-high speed high capacity transportation, [1]

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Figs. 1 to 4, pp. 370 to 377.

Object of the present paper. — As a result of the paper we recently read before the Académie des Sciences (2), in which certain people have thought to see an invention to some extent connected with the various projects of « guided flight » brought forward during recent years, we feel it is indispensable to resume below the principal technical data on which our investigation was based.

It will thereby be seen that without making use of methods entirely outside the railway sphere, such as the use of suspension and balancing planes, or even air screws, and without pretending at the start to deal with the very complicated problem of the ultra-rapid conveyance of heavy and bulky loads, we have limited ourselves to a precise statement of what can be done to-day as regards light railways adapted for the ordinary movement *on rails* of automotors of small size and light weight of carefully designed aerodynamic form capable of running at speeds which have become industrially available.

At the moment when the attention of economic organisations is greatly excited by the question of the acceleration of the postal services, it appears to us opportune to underline, in this respect, the remarkable possibilities of speed and capacity in railways when adapted to the proposed extra light transport.

Furthermore, the only points that could be in question were the fundamental principles of the formula which in the present state of industrial knowledge seemed to us the most simple and the most economical

(1) Translated from the French.

(2) Proceedings of the *Académie des Sciences*. Meeting of the 3 January 1928. Gauthier-Villars & Co., Publishers, Paris.

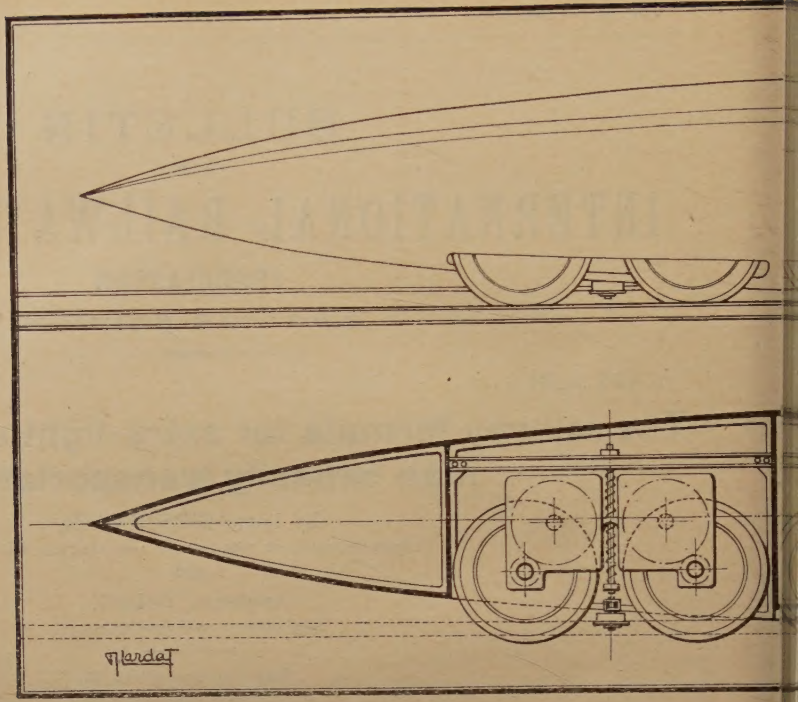


Fig. 1. — Diagram

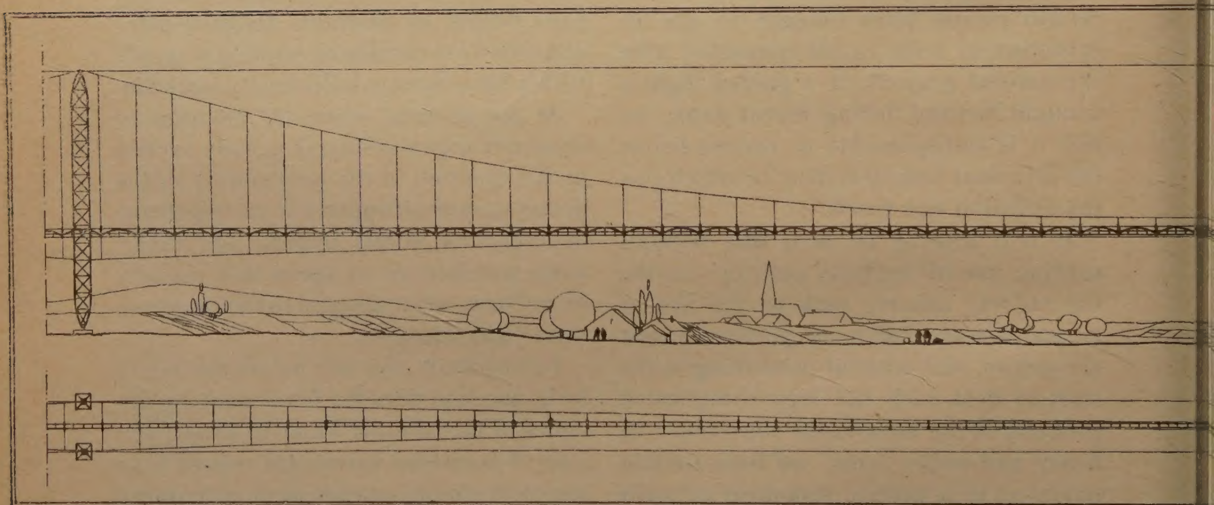
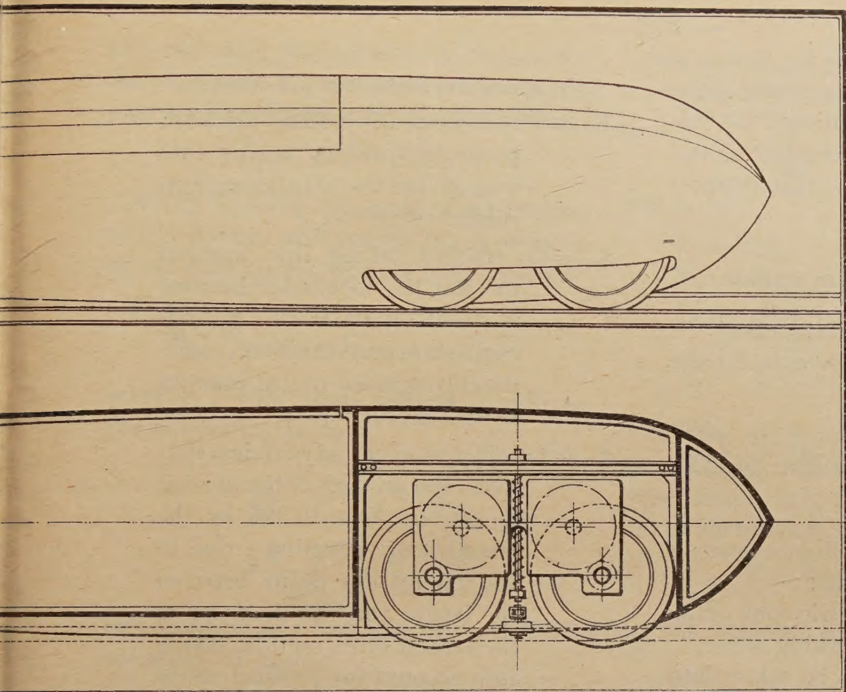
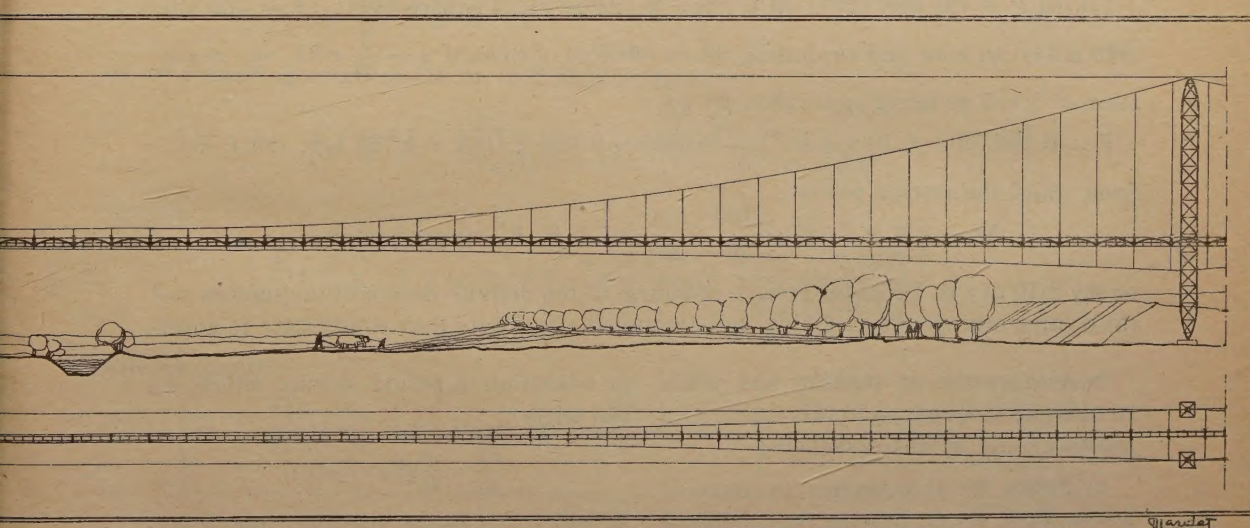


Fig. 2. —



ultra-high speed automotor.



of a typical span.

(Drawings by R. Lardat.)

to utilise : it is advisable in consequence to take the numerical quantities of the calculations as giving an indication of the order of magnitude only so that definite ideas may be formed.

I. — The ultra-rapid locomotor.

1° Calculation of the tractive effort. — We have used the general formula for the resistance to movement [1] :

$$F = P \cdot \left[K_f + K_r \cdot \left(\frac{V}{100}\right)^{1.65}\right] + K_a \cdot S \cdot \left(\frac{V}{3.6}\right)^2$$

in which

- F = Tractive effort in kilogrammes on straight level road;
- P = Total weight in metric tons of the locomotor in working order;
- S = Frontal surface to be taken into account in view of the aerodynamical resistance (maximum diameter of 0.35 to 0.40);
- V = Speed in kilometres per hour;
- K_f = Co efficient of rolling friction, properly speaking taking into account the use of ball bearings, rubber tyres, etc.;
- K_r = Co efficient taking into account the way the locomotor behaves at speed on its track : rolling, oscillation and vibrations considerably damped out in practice by the rubber tyres;
- K_a = Co efficient of aero-dynamic resistance, the product of the normal co efficient $K = 0.085$ by the constant of reduction ρ due to the elongation (ratio between the length and the diameter at the largest cross section) taking into account the profiled shape of the body and the perfect smoothness of the outside surface.

Taking $P = 125$ kgr. (275.6 lb.); $S = 0.125$ m² (1.34 square feet); $V = 360$ km. (225 miles) an hour and supposing the co efficient of Renard $\rho = \frac{1}{31}$ with an elongation of 5 to 7 as being applicable, we get :

$$F = 0.125(10 + 0.10 \times \overline{3.6}^{1.65}) + 0.0027 \times 0.125 \times \overline{100}^2 = 4.725 \text{ kgr. } (10.4 \text{ lb})$$

from which the average power :

$$W = 100 \times 4.725 \times \frac{100}{85} \times \frac{1}{75} = 7.5 \text{ H. P. approximately,}$$

or say 8/10 of a H. P. under reserve of course of the definite design of the motors and the cooling arrangements, taking into account the real duty they would have to cover.

2° Acceleration at starting and when running up to speed. — In this matter two phases should be distinguished :

a) Period of acceleration at constant power, $F_a \leqslant P \cdot f_a$, f_a being the coefficient of adhesion, a period during which we can write :

$$\frac{dv}{dt} = \frac{g}{P}[F_a - \varphi(v)],$$

in which g is the acceleration due to gra-

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[1] Cfr. A. Talon : Paper before the *Académie des Sciences* of the 29 December 1924.

vity and $\varphi(v)$ the general expression for the resistance to movement which can in practice be reduced to the quadratic form:

$$\varphi(v) = \alpha + \beta V^2.$$

When V_a being the speed of application of full power, we have for the expression of the time corresponding to this period :

$$\frac{g}{P} t_a = -\int_0^{V_a} \frac{dv}{F_a - \alpha - \beta \cdot V^2}$$

whence, by the usual method of solving the quadratic, taking :

$$\sqrt{\frac{\beta}{F_a - \alpha}} = \mu,$$

$$t_a = \frac{P \cdot \mu}{2g\beta} \cdot L \frac{1 + \mu V_a}{1 - \mu V_a}.$$

We should get in the same way for the corresponding distance :

$$a = \frac{P}{2g\beta} L \frac{1}{1 - \mu^2 \cdot V_a^2},$$

that is for :

$P = 125$	$\alpha = 1.25$
$F_a = 20$	$\beta = 0.0003475$
$V_a = 25.5$	$\mu = 0.043$
$g = 9.81$	$a = 222$ m.
$t_a = 17.2$ s.	

b) Time of acceleration at full power, time during which the effort F_p varies either simply inversely as the speed according to the formula :

$$F_p \cdot V = W \times 75 \times \frac{85}{100} = K,$$

or according to a formula, such as a parabola for example, taking into account the variation of output.

In the first case the formula of the times will be, V_m being the real speed :

$$\frac{g}{P} \cdot t_p = \int_{V_a}^{V_m} \frac{dv}{\frac{K}{V} - \alpha - \beta V^2} = -\int_{V_a}^{V_m} \frac{V \cdot dv}{\beta V^3 + \alpha V - K}$$

or for a limiting speed of 110 m. (361 feet) a second :

$$\frac{g}{P} \cdot t_p = -\int_{V_a}^{V_m} \frac{V \cdot dv}{\beta (V - 110)\left[(V + 55)^2 + \overline{114.8}^2\right]}$$

whence :

$$\overline{114.8}^2 \frac{g}{P} \cdot \beta \cdot t_p = -\int_{V_a}^{V_m} \frac{35.9\, dv}{V - 110} + \frac{0.395 - 0.0027\, V}{1 + \left(\frac{V + 55}{114.8}\right)^2}$$

whence finally :

$$t_p = P \left\{ \frac{35.9\, L \frac{110 - V_a}{110 - V_m} - 62.45\left(\text{arc tg} \frac{V_m + 55}{114.8} - \text{arc tg} \frac{V_a + 55}{114.8}\right) + 17.79\, L \frac{1 + \left(\frac{V_m + 55}{114.8}\right)^2}{1 + \left(\frac{V_a + 55}{114.8}\right)^2}}{\overline{114.8}^2 \cdot \beta \cdot g.} \right.$$

In the same way for the corresponding distance :

$$e_p = \mathrm{P}\left\{\frac{3.949 \cdot \mathrm{L}\dfrac{110-\mathrm{V}_a}{110-\mathrm{V}_m} - 567\left(\operatorname{arc\,tg}\dfrac{\mathrm{V}_m+55}{114.8} - \operatorname{arc\,tg}\dfrac{\mathrm{V}_a+55}{114.8}\right) - 4\,614\,\mathrm{L}\dfrac{1+\left(\dfrac{\mathrm{V}_m+55}{114.8}\right)^2}{1+\left(\dfrac{\mathrm{V}_a+55}{114.8}\right)^2}}{114.8^2 \cdot \beta \cdot g.}\right.$$

whence for $\mathrm{V}_m = 100$, *i. e.*, 360 km. (225 miles) an hour :

$$t_p = 186 \text{ s.} \qquad e_p = 14\,638 \text{ m.}$$

and in total :

$$\mathrm{T} = t_a + t_p = 203 \text{ s.} \qquad \mathrm{E} = e_a + e_p = 14\,860 \text{ m. } (48\,754 \text{ feet}).$$

3º **Automatic safety braking.** — The question of safety braking, coupled to the automatic interruption of the supply to the line is most important to consider on account of the exceptionally high figures taken for the speeds which affect the calculation of the inertia to be absorbed, but it does not present any particular difficulty : in the case of electro-magnetic braking on the rail, the length of stop being taken as 2 000 m. (6 562 feet) the longitudinal effort taken by the aerial line would not exceed :

$$\frac{1}{2} \times \frac{125}{9.81} \times 100^2 \times \frac{1}{2\,000},$$

or about 35 kgr. (77 lb.), which is not in any way excessive.

4º **Effect of the wind.** — A head wind of V′ km. an hour would reduce the available power by $0.003 \times 0.125\left(\frac{\mathrm{V}'}{3.6}\right)^2$. So that for $\mathrm{V}' = 120$ the maximun speed $\mathrm{V}_m = 400$ (250 miles) corresponding to the effective local used power of 10 H. P. would be reduced to $400 \times \frac{6 - 0.408}{6} = 372$ km. (231 miles) an hour, taking into account the fact that a wind of this intensity does not blow continuously but in gusts of varying intensity. For a similar side wind at an acute angle to the track of γ the adverse speed would obviously be $\mathrm{V}' \cos \gamma$.

As for the side thrust, it would be a maximum when $\gamma = 90°$ and with a wind of 72 kgr. per square metre (14.75 lb. per square foot) of the longitudinal section of the fish shaped body suggested (adopted by analogy from the maximum laid down in the Ministerial decision of the 30 April 1927 on overhead electric power lines). This thrust of the order of 50 kgr. (110 lb.) would then be without appreciable influence either on the rolling resistance [increase of tractive effort less than 0.25 kgr. (0.55 lb)] or on the mechanical reaction of the superstructure properly stayed as we shall see in the transverse plane.

These are the general figures for the traction by the ultra-rapid automotor considered, having a power of 8/10 H. P. which would be ample to rely upon under all circumstances on a section with few gradients, for us to be able to count on commercial speeds of the order of 360 km. (225 miles) an hour.

II — The aerial track.

1° Calculation of the suspended overhead structure. — The general formulæ of T. Godard are to be used here, the case of very long spans, of very flattened out arch, having a stiffening girder taken as being held in above the supports, only having to support the passage of light individual weights, which when crossing can cause a maximum localised additional load of 250 kgr. (550 lb.).

The notation adopted is as follows :

l = the length of the girder between bearings,

p = the permanent load per linear metre : cables, girders and suspenders,

P = travelling load,

λ = the distance from this concentrated overload to the left-hand end,

λ' = the distance from this concentrated overload to the right-hand end,

N = the horizontal component of the tension in the cables under the action of the permanent load,

ΔN = the increase in this component due to the action of the overload P,

K = a constant given by the expression $\sqrt{\frac{N + \Delta N}{EI}}$ in which I is the moment of inertia of the section of the girder and E the co-efficient of elasticity of the metal, ΔN being very small in relation tot N, K has the approximative value $\sqrt{\frac{N}{EI}}$.

δ = the coefficient of expansion of the steel,

τ = the variation of temperature in regard to the temperature of construction,

R_1 = the reaction over the left-hand support,

R_2 = the reaction over the right-hand support,

μ_1 = the bending moment at the left-hand support,

μ_2 = the bending moment at the right-hand support,

d = the spacing of the suspenders,

Δ_p = the unit increase in tension in the suspenders due to the action of the moving load,

Δ_p is given as

$$\Delta_p = p_1 e^{kx} + p_2 e^{-kx}$$

or

$$\Delta_p = p'_1 e^{kx} + p'_2 e^{-kx},$$

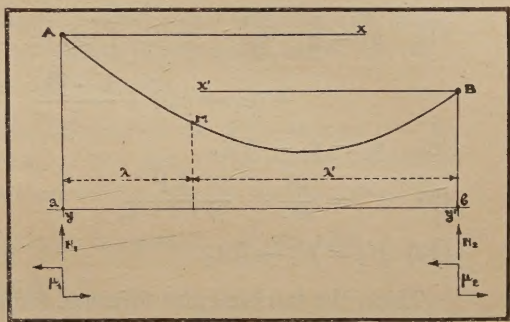

Fig. 3.

according as (fig. 3) the coordinates x, y, of the superstructure are considered in relation to one or other of the systems of axes Ax, Ay, Bx', By', passing respectively by one or other of the two ends A and B of the cable : p_1, p_2, p'_1, and p'_2 are constants depending on λ and λ'.

Taking also, with e the base of the logarithms :

$$a = \frac{e^{kl} + 1}{e^{kl} - 1}, \quad \alpha = \frac{e^{k\lambda} + e^{k\lambda'}}{e^{kl} - 1},$$

$$\beta = \frac{e^{k\lambda'} - e^{kl}}{e^{kl} - 1}.$$

A) Influence of the rolling loads. — The formulæ are the following :

(1) $$N = \frac{p \cdot l^2}{8f}.$$

(2) $$\frac{\Delta N}{N} = \frac{P}{pl} \cdot \frac{K^2\lambda\lambda' - Kl(a-\alpha)}{2 - K \cdot l \cdot a + \frac{K^2 \cdot l^2}{6}}.$$

(3) $$\mu_1 + \mu_2 = \frac{P\lambda\lambda'}{l} - p \cdot \frac{l^2}{6} \cdot \frac{\Delta N}{N}.$$

(4) $$\mu_1 - \mu_2 = P \cdot \frac{l\beta + \lambda - \lambda'}{2 - K \cdot l \cdot a}.$$

(5) $$R_1 = \frac{P\lambda'}{l} - \frac{pl}{2} \cdot \frac{\Delta N}{N} + \frac{\mu_1 - \mu_2}{l}$$

(6) $$R_2 = \frac{P\lambda}{l} - \frac{pl}{2} = \frac{\Delta N}{N} - \frac{\mu_1 - \mu_2}{l}$$

(7) $$p_1 = \frac{p}{2} \cdot \frac{\Delta N}{N} - \frac{K^2}{2}[(\mu_1 + \mu_2) + + (\mu_1 - \mu_2)] + \frac{K \cdot R_1}{2}$$

(8) $$p_2 = p_1 - K R_1$$

(9) $$p'_1 = \frac{p^2}{eKl} + \frac{KP}{2eK(l-\lambda)}$$

(10) $$p'_2 = p'_1 - KR_2.$$

Then, the fundamental formulæ for the bending moment and the shear T deduced from the preceding one given for a section of abscissa x :

(11) $$M = \frac{p_1 e^{kx} + p_2 e^{-kx}}{K_2} - \frac{p}{K_2} \cdot \frac{\Delta N}{N}$$

(12) $$T = \frac{dM}{dx} = \frac{p_1 e^{kx} - p_2 e^{-kx}}{K}$$

B) Influence of the temperature. — The formulæ are the following :

(1^{bis}) $$\frac{\Delta N}{N} = \frac{K^2 \cdot N^2}{p^2} \times \frac{\delta\tau}{\frac{Kla}{2} - K^2\left(\frac{l^2}{12} + \frac{N^2}{p^2}\delta\tau\right) - 1}$$

(2^{bis}) $$p_1 = \frac{Kl}{2} p \frac{\Delta N}{N} \cdot \frac{l}{e^{kl} - 1}$$

(3^{bis}) $$p_2 = p_1 \cdot e^{kl}$$

(4^{bis}) $$\Delta p = \frac{p}{2} \cdot \frac{\Delta N}{N} \cdot \frac{Kl}{e^{kl} - 1}(e^{kx} + e^{kx'})$$

M and T retaining the same form as above but with new values of p_1 and p_2.

It should be noted, however, that taking into account, on the one hand, the use of metal towers, and on the other, the semi-rigidity of the structure considered, the effects of the variations of the temperature would be inappreciable as regards the increase in fatigue of the parts of the overhead structure. We shall therefore take no notice of it in our calculations of the strength, but we shall see later on what effect it has on the thermic deformation of the longitudinal profile of the line.

2. **Calculation of the details of the overhead structure.** — Let us consider, as a first indication, a typical span of 500 m. (1 640 feet) with 25 m. (82 feet) sag, and let us suppose the supports at the same level, that is to say, that the overhead structure be symetric as regards a normal plane passing through the centre of its span.

a) Calculations of the booms of the girder. — All that is required is to study the variations in the maximum bending moments calculated in the various sections of the girder caused by the live load moving in the first left-hand half of the span.

When this live load is at 1/10th of the span, *i. e.*, at 50 m. (164 feet) from the left support, the maximum bending moment is equal to

$$0.0245 \times 500 \times 250 = 615 \text{ Kg. M.}$$
(4 448 foot-pounds);

When it is at 100 m. (328 feet) this maxi-

mum moment is 4 465 Kg M. (32 295 foot-pounds).

When it is at 150 m. (492 feet) this maximum moment is 4 588 Kg. M. (33 185 foot-pounds), and when it is at 200 m. (656 feet) this moment is reduced to 4 265 Kg. M. (30 850 foot-pounds).

From the curve plotted through these maximum bending moments (fig. 4) we take as maximorum value

M = 4 600 Kg. M. (33 270 foot-pounds).

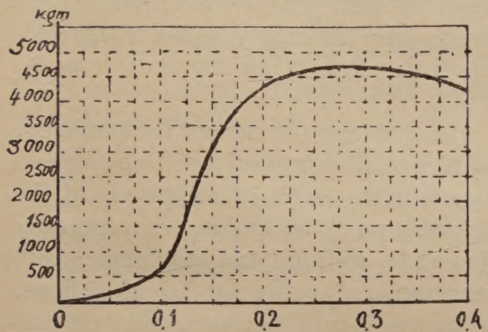

Fig. 4.

To arrive at M we haven taken a coefficient of rigidity:

$$Kl = l\sqrt{\frac{N}{EI}} = 10,$$

as in the case of modern suspension bridges of long span, so that as a first approximation p = 25 kgr. (55 lb.), whence :

$$N = \frac{p \cdot l^2}{8f} = 31\,250 \text{ kgr. } (68\,890 \text{ lb.}).$$

The moment of inertia of the section of the run of the girder then becomes

$$I = \frac{l^2 \cdot N}{100E} = 0.0031.$$

Furthermore, if h is the depth of the girder and Ω the area of the booms, when taking $\Omega \times 7\,800 = 12$ kgr. (26.4 lb.), whence $\Omega = 0.0016$, we shall have approximately from the usual formula :

$$\Omega\,\frac{h^2}{4} = I,$$

$$h = 2\sqrt{\frac{0.0031}{0.0016}} = 2 \times 1.40 \text{ m.}$$

As regards the maximum stress in the booms, it would be given by

$$\frac{M}{\frac{I}{V}} = \frac{4\,600 \times 1.40}{0.0031},$$ that is barely 2 kgr. per square millimetre (1.27 English tons per square inch), which gives an ample margin of safety whatever accidental additional loading, due to gales, snow, ice, etc., have to be covered.

b) Calculation of wind bracing. — It will meet the case if the maximum shear stresses arising near the central part of the span are calculated. If we take $\lambda = \frac{l}{2}$, we get :

$$T = \frac{p_1 e^{\frac{Kl}{2}} - p_2 e^{-\frac{Kl}{2}}}{K} = 25\,(p_1 e^5 - p_2 e^{-5}),$$

with $p_1 = 0.034$ and $p_2 = 3.764$, whence T = 125 kgr., which gives a maximorum value in shear of about 63 kgr. (138.9 lb.) per upright.

The minimum section of the T bars of the uprights being 60 square millimetres (0.09 square inch) we see that the stress which amounts to about 1 kgr. per square millimetre (0.635 English ton per square inch) taking into account, if need be, of the coefficient of excentric loading in the parts in compression, also leaves a considerable margin to meet all emergencies.

c) Calculation of the suspenders. — The unit increase in tensile stress Δ_p, carried

by a suspender, as the result of the passage of the moving load, has its maximum maximorum value at the middle of the span where we get :

$$\Delta p = p_1 e^5 + p_2 e^{-5} = 2 \text{ kgr. } 525 (5.57 \text{ lb.})$$

The suspenders being taken as spaced 5 m. (16 ft. 5 in.) apart, the assembly of two suspenders corresponding to one single section, will therefore carry in all a load equal to about 138 kgr. (304 lb.), that is, 70 kgr. (154 lb.) each. Piano wire of extra high tensile steel of 4 mm. (5/32 inch) diameter will therefore be amply large enough.

d) Retaining cables and top cables. — If a represents half the transverse spacing of the columns such that the transverse stability of the axial girder is always assured by the angle of the suspension itself even under the thrust due to gales of 120 kgr. (265 lb.), we ought to get approximately :

$a = 120 s \cdot \frac{f}{p_g}$, s being the maximum surface acted on by the wind, f the deflection and p_g the permanent unit weight of the suspended superstructure :

whence for $s = 0.20$ m² (2.22 square feet) $f = 25$ m. (82 feet) $p_g = 35$ kgr. (77 lb.),

$$a = 17 \text{ m. } (55.77 \text{ feet}).$$

For a spacing $a' < a$, the increase of thrust p_v to be balanced out especially by the retaining cable would be under the mentioned conditions mentioned above,

$$p_v = 42 \times \frac{2.65 - 0.15\, a'}{\sqrt{25^2 + a'^2}},$$

or for $a' = 10$, for exemple, $p_v = 1$ kgr. 800 (3.96 lb.) which would correspond to two retaining cables weighing about 1 kgr. per linear metre (2 lb. per yard).

To calculate the *top cables*, all that is needed is to obtain the value of the maximum tension to be balanced, which for a span of 400 metres (1 312 feet) would be equal to [1] :

$$\frac{250}{2} \times \frac{400}{4} \times \frac{1}{25} = 500 \text{ kgr. } (1\,100 \text{ lb.})$$

The top cables therefore of the adjacent section will have the total cross section $\sigma = 20$ square millimetres (0.031 square inch) in steel wire — but there is no reason why the supply feeder of the electric trolley should not also do this work.

e) Structural and thermic Deformations. — It is of the greatest use to be able to ascertain the order of magnitude of the deformation of the profile of the running line under rolling loads and temperature variations. For structural deformations, let us consider in the case of the type examined the expression for the length L of the suspension cables at the temperature when constructed :

$$L = l\left(1 + \frac{8}{3}\frac{f^2}{l^2} - \frac{32}{5} \times \frac{f^4}{l^4}\right),$$

whence L = 503.313 m. (1 651.319 feet).

Let us first of all imagine the girder as having no rigidity. The effect of the concentrated live load P is to cause for each position of this load a deformation easily calculated at points by means of Résal's equations, which we think it unnecessary to reproduce here.

In particular, the maximum deformation

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(1) Cfr. Leinekugel Le Cocq. Note to the Académie des Sciences.

which occurs when the moving load is at the middle of the span can be deduced from the following equation in which f' is the new ordinate :

$$L = l + \frac{2}{3} f^{2}_{m} \frac{2\left(\frac{l}{2} + \frac{P}{2p}\right)^3 - \frac{P^3}{4p^3}}{\left[\left(\frac{l}{2} + \frac{P}{2p}\right)^2 - \frac{P^2}{4p^2}\right]^2} - \frac{2}{5} f^{4}_{m} \frac{2\left(\frac{l}{2} + \frac{P}{2p}\right)^5 - \frac{P^5}{16 - p^5}}{\left[\left(\frac{l}{2} + \frac{P}{2p}\right)^2 - \frac{P^2}{4p^2}\right]^4}$$

from which we deduce $f' = 25.238$ m. (82.803 feet).

The effect of the rigidity of the girder is seen in a reduction of the deformation, approximately equal, according to a note of Mr. Pigeaud to $\frac{M}{N + \Delta N}$, whence we get at the middle of the span a structural deformation :

$f_m = f' - 0.14 = 25.10$ m. (82.35 feet) that is to say, a maximum increase in gradient of the longitudinal section of less than 1/2 millimetre per metre (1 in 2 000) which is practically negligible.

The variations in deflections through temperature changes from the temperature of construction are given approximately by the formula :

$$\delta = \frac{15}{80} \times \frac{l^2}{f} \cdot \alpha \cdot t,$$

α being the coefficient of linear expansion of the steel of the cables. For $t = 25^\circ$ for example, $\delta = 0.52$ m. (1.7 feet) or a maximum increase in gradient on the longitudinal section of 2 millimetres per metre (1 in 500) which is not excessive even on ultra high speed railways.

It must however be noted that the deformation due to temperature changes varies inversely as the square of the span, so that if the length of the standard span were made 400 m. (1 312 feet) the maximum lowering of the level at the centre would be reduced to $0.52 \times \frac{16}{25} = 0.33$ m. (1.08 feet) the maximum variation in gradient being in consequence brought down to only about 1 millimetre per metre (1 in 1 000).

f) THE TOWERS. — The towers would be of an uniform type [1] built up of four corner members formed of 60 × 60 × 6 (2.362 × 2.362 × 0.236 inch) L bars weighing 5.275 kgr. a metre (10.63 lb. per yard) with stays of 60 × 60 × 6 (2.362 × 2.362 × 0.236 inch) L bars stays having 60 × 6 (2.362 × 0.236 inch) flat cross braces.

The maximum load in compression load in the case of two panels 600 m. (1 968 ft. 6 in.) panels being about 12 500 kgr. (27.560 lb.), the maximum constraining resistance required of the corner members will be :

$$\frac{12\,500}{\frac{4 \times 5.275}{0.0078}} = 4.1 \text{ kgr. per square millimetre (2.6 tons per square inch).}$$

which gives a factor of safety of over *10*.

The weight of each tower 35 m. (114.8 feet) high would be about 2 500 kgr. or $\frac{5\,000}{500} = 10$ kgr. per metre run (20 lb. per yard) of the overhead structure.

Cost of the overhead structure. — Based

(1) A design like that due to Mr. Leinekugel Le Cocq, Engineer, suspension bridge builder.

upon the above figures, which are obviously minima, the aerial overhead structure considered would include, per metre (per yard) run :

1. Rolled steel (rigid girder with running rails) about 15 kgr. (30 lb.); towers 10 kgr. (20 lb.) or in all 25 kgr. (50 lb.).

2. Cables of high tensile steel with a working strength of 35 kgr. per square millimetre (22.22 tons per square inch), an elastic limit of 70 kgr. (44.45 tons per square inch) and a breaking strength of 120 kgr. (76 19 tons per square inch) weighing 8 kgr. per metre (16 lb. per yard) including suspenders.

We may take the price of rolled material at 115 francs the 100 kgr. the basic price, which is definitely higher than anything probable at present.

The cost of manufacture, in view of the standardisation of the details of the metal constructional work, can be taken as equal to 60 francs, and the cost of erection at 45 francs per 100 kgr. The constructional work will therefore cost in all 25 × 2.20 fr. = 53 francs per linear metre of line (16. 15 fr. per linear foot).

The wire used in making the suspension cables can be valued at 185 francs the 100 kgr. and the cables at 340 francs the 100 kgr. in position, that is to say, 25 francs per linear metre of line (7.61 fr. per linear foot).

To sum up the cost per linear metre of the metal, aerial overhead steel structure would be about 80 francs (24.38 fr. per linear foot). If to this estimate be added the cost of the electric feeder line and the masonry foundation of the towers, it will be seen that *even with an additional 25 % in the weight of the steel work provided for* the total cost price would only vary between 100 to 120 000 francs per kilometre (160 000 and 193 000 fr. per mile).

III. — Operating figures.

Particulars and calculation of cost. — Let us take :

p_r = the cost price of carriage per useful metric ton-kilometre,

K_u = the coefficient of useful load, that is to say, the ratio of the useful tonnage C_u, to the gross tonnage, $C_u + P_m$, P_m being the tare or dead weight,

K_a = the coefficient of aero-dynamic resistance,

K_ρ = the coefficient of mechanical efficiency of the locomotor,

K_m = the coefficient of potential energy of the locomotor,

$\overline{\omega}$ = cost of the horse power hour,

L_r = the actual length of the line,

L_v = the virtual length of the line, including the increases due to the gradients and curves,

V = average running speed applicable to L_v,

K_i = the coefficient of immobilisation of the rolling stock,

θ = the lay-off time in hours between two trips of the same locomotor,

T = the gross tonnage hauled per trip,

Q_j = useful daily tonnage,

p_a = the coefficient of amortization of the motor,

p_e = the coefficient of annual maintenance of the motor,

p'_a = the coefficient of amortization of the rolling stock,

p'_e = the coefficient of annual maintenance of the rolling stock,

p''_a = the coefficient of amortization of the fixed plant,

p''_e = the coefficient of annual maintenance of the fixed plant,

K_g = the coefficient of capital charges,

λ = the minimum headway,

$I = \frac{60\lambda}{V}$, the headway interval in minutes,

N = the number of automotors in service,

$\varphi(v) = K_f + K_r \cdot V^{1.65} + 8K_a \cdot S \cdot V^2$,
= the tractive effort in kilogrammes per gross metric ton at the speed of V km. an hour,

$W = T \cdot V \cdot \varphi(v) \cdot \frac{1}{K\rho} \cdot \frac{1}{75}$, the horse power of the locomotor.

In the various practical applications it will be supposed the useful tonnage *offered* is 25 kgr. (55 lb.) for each automotor following another at regular intervals, that is, for example, at the maximum rate $I = \frac{60\lambda}{V}$ for the four busiest hours of the day of the period between 9 to 20 hours, at 5 minute intervals for the other seven hours of the same period, and at a rate of 15 minutes between 6 and 9 hours and from 20 to 24 hours.

Under these conditions :

1. The power consumed in relation to the movement of the dead weight of the automotors remains constant for a given speed.

2. The total number N of automotors included in the rolling stock park ought to be fixed in accordance with the useful maximum output.

In this way wet get :

a) The daily number of single journeys :

$$N = 2\left(\frac{4V}{\lambda} + 112\right)$$

b) Horse power hour used daily :

$$C_j = (N \times 0.100 + Q_j)\,\varphi(V) \cdot L_v \frac{1}{K\rho}\frac{1}{75} \times \frac{1}{3.6}$$

c) Maximum useful tonnage carried per automotor per hour :

$$T_m = \frac{V \cdot T \cdot K_u}{L_v + \theta V}(1 - K_i)$$

d) Maximum hourly tonnage :

$$Q_m = \frac{60T \cdot K_u}{I} = \frac{V \cdot T \cdot K_u}{\lambda}$$

e) Number of automotors in the stock :

$$N = (L_v + \theta V) \cdot \frac{1}{(1 - K_i)}$$

f) Amortization and daily maintenance of the motor stock :

$$A_m = N \cdot W \cdot K_m \cdot \frac{p_a + p_e}{365}$$

g) Amortization and daily maintenance of rolling stock :

$$A_r = N(T - W \cdot K_m) \cdot \frac{p'_a + p'_e}{365}$$

h) Amortization and daily maintenance cost of the fixed plant :

$$A_f = L_r \cdot \frac{p''_a + p''_e}{365}$$

whence, finally :

$$p_r = \frac{1}{Q_j \cdot L_r}\left(C_j\varpi + (1 + K_g)(A_m + A_r + A_f)\right)$$

or developed, giving the coefficients the following values :

$$K = 0.8,\quad K_m = 10,\quad K_i = 0.10,\quad K_g = 0.10,$$

$$p_a = 15,\quad p_e = 10,\quad p'_a = 10,\quad p'_e = 5,\quad p''_a = 8\,000,\quad p''_e = 2\,000,$$

$$\varpi = 0\,60,\quad \lambda = 4,\quad \theta = 0.1,\quad \frac{L_v}{L_r} = 1.25,$$

$$p_r = \frac{1}{Q_j}\left[32.10 + 0.000347\varphi(V)\left(10Q_j + 2.137V + 0.011\frac{V^2}{L_r} + 224\right) + 0.0157\frac{V}{L_r}\right].$$

This formula now has only three variables : the daily useful output Q_j, the average running speed V and the length of the line L_r; but this third factor does not come into question, as it were, as regards the two essential elements of the cost price : the amortization and the kilometric cost of maintenance of the line and the daily consumption of power per kilometre corresponding to a given output.

Influence of the speed V. — If the speed be taken as the only variable, the condition $\frac{\delta p_r}{\delta V} = 0$ theoretically allows us to determine the speed of economical working corresponding to the maximum output, but this consideration of the economical speed becomes secondary in the cost of extra light transportation in which the matter carried (letters, securities, luxury goods, etc.) can bear the increase in cost due to the increase in speed which it is always advantageous to make the maximum practicable.

Influence of the daily tonnage Q_j. — This factor must appear in the formula for the cost price, and it would obviously be of value, from the economical point of view, to make the output the maximum allowed by the headway formula. However, this factor of limited output is not of first importance when the insignificant effect of the cost price per metric ton-kilometre on the unit (usually the gramme) of the material carried is considered.

Influence of the first cost of the line. — This influence becomes the greater, the smaller the daily traffic handled. If the output were so small as to be covered by outward and return trips it would be possible to use one line only, the coefficient p''_a et p''_e would then be reduced by about as much as 3/8 and 1/4 respectively. But the essential value of the ultra-high speed method of transportation considered lies above all in the regularity and volume of traffic handled, especially during the busiest hours. Operating with two lines is the only possible method under such conditions.

IV. — Data for an application.

It is evident, *a priori*, that the proposed system cannot and ought not to be considered except in countries where there exists over relatively short distances, heavy commercial traffic — business, banking, etc., letters or papers — and can only be justified between centres of dense population such as is the case of the large cities of Western Europe. In other cases, air transport, properly speaking, will obviously be more suitable.

Table 1 below sums up the information as to the application of the formulæ of cost price to the case of the working of a line 400 km. (250 miles) long, operated according to the various hypotheses considered in column 1 of the said table.

TABLE 1.

Cost price of transportation per metric ton-kilometre by aerial railway according to the ratio of the traffic carried to that offered.

USEFUL DAILY TONNAGE.	Number of return journeys.	Total traffic offered in metric tons.	Percentage used.	Cost price per metric ton-kilometre. V = 360 km. (225 miles) an hour.	V = 240 km. (150 miles) an hour.	V = 120 km. (75 miles) an hour.
				Francs.	Francs.	Francs.
Q_j = 0.500 t. in each direction.	472	24	4.2	44.67	...	...
	352	18	5.6	...	36.27	...
	232	12	8.4	...	...	32.80
Q_j = 1 t. in each direction. . .	472	24	8.4	22.40	...	...
	352	18	11.2	...	18.18	. .
	232	12	16.8	...	...	16.40
Q_j = 5 t. in each direction. . .	472	24	42	4.58	...	...
	352	18	56	...	3.68	...
	232	12	84	...	...	3.29
Q_m = Output at full load. . . .	472	24	100	1.98	. .	. .
	352	18	100	...	2.07	. .
	232	12	100	...	...	2.74
Q_M = Maximorum output . . .	1 440	72	100	0.75	..	...
	960	48	100	...	0.81	...
	480	24	100	...	...	1.38

Table 2 below gives concrete cost prices applied to the carriage over a distance of 400 km. (250 miles) of a letter weighing 20 gr. (0.7 oz.) and a parcel weighing 1 kgr. (2.2 lb.).

TABLE 2.

Cost price of transportation over 400 km. (250 miles) at a speed of 360 km. (225 miles) an hour of a letter weighing 20 gr. (0.7 oz.) and a parcel weighing 1 kgr. (2.2 lb.) under various percentages of use of the traffic offered varying between 1 and 100 %.

USEFUL DAILY TONNAGE.	Percentage used.	Cost price of carriage for a distance of 400 km. (250 miles): of the letter weighing 20 gr. (0.7 oz.)	of the parcel weighing 1 kgr. (2.2 lb.)	*Remarks.*
		Francs.	Francs.	
Q_j = 0.120 t. in each direction.	1	1.49	74.29	
Q_j = 0.500 t. — — .	4.2	0.36	17.87	Daily receipts : 17 800 francs.
Q_j = 1 t. — — .	8.4	0.18	8.96	Annual receipts : 6 500 000 francs.
Q_j = 2 t. — — .	16.8	0.09	4.50	Average receipts for a single journey : 18.80 francs.
Q_j = 5 t. — — .	42	0.037	1.83	
Q_m = 12 t. — — .	100	0.016	0.79	
Q_M = 36 t. — — .	100	0.006	0.30	

V. — Comparison with other methods of transport.

For comparative purposes, it will be of interest to give in tables 3 and 4 below some particulars as to the cost price of the various methods of transportation, which could be used in concrete cases analogous to those considered above.

TABLE 3.

Determination of the cost price of automobile transport for a distance of 400 km. (250 miles) at a speed of about 75 km. (47 miles) an hour.

DAILY USEFUL TONNAGE [1].	Coefficient of use.	Cost price		Remarks.
		per kilometre.	per metric ton-kilometre.	
1. Motors of 1.1 to 1.5 litres cylinder volume. — Useful load : 125 kgr. (275 lb.).				
Q_j = 0.080 t. in each direction.	31 %	1.70 fr.	42.40 fr.	[1] The cost price is independent of the frequency.
Q_j = 0.160 t. — — .	62 %	1.70 fr.	21.20 fr.	
Q_j = 0.250 t. — — .	100 %	1.70 fr.	13.60 fr.	
2. Motors of 2.5 to 3 litres cylinder volume. — Useful load : 400 kgr. (880 lb.).				
Q_j = 0.250 t. in each direction.	31 %	2.50 fr.	20.00 fr.	
Q_j = 0.500 t. — — .	62 %	2.50 fr.	10.00 fr.	
Q_j = 0.800 t. — — .	100 %	2.50 fr.	6.25 fr.	

TABLE 4.

Determination of the cost price of air transport over a distance of 400 km. (250 miles) at an average route speed of 160 km. (100 miles) an hour.

DAILY USEFUL TONNAGE.	Percentage utilised.	Number of return journeys.	Cost price	
			per flying hour.	per metric ton-kilometre.
1. Flying machines of 100 H. P. with a coefficient of useful load of 90/750 kgr. (200/1 650 lb.).				
Q_j = 0.045 t. in each direction	25	2	421 fr.	116 00 fr.
Q_j = 0.090 t. — —	25	4	390 fr.	108.40 fr.
Q_j = 0.090 t. — —	50	2	421 fr.	58.00 fr.
Q_j = 0.180 t. — —	50	4	390 fr.	54.20 fr.
Q_m = 0.180 t. — —	100	2	421 fr.	29.00 fr.
Q_m = 0.360 t. — —	100	4	390 fr.	27.10 fr.

TABLE 4 (*continued.*)

DAILY USEFUL TONNAGE.	Percentage utilised.	Number of return journeys.	Cost price.	
			per flying hour.	per metric ton-kilometre.
2. Flying machines of 230 H. P. with a coefficient of useful load of 500/1 750 kgr. (1 100/3 680 lb.).				
Q_j = 0.250 t. in each direction	25	2	717 fr.	36.00 fr.
Q_j = 0.500 t. — —	25	4	670 fr.	33.60 fr.
Q_j = 0.500 t. — —	50	2	717 fr.	18.00 fr.
Q_j = 1.000 t. — —	50	4	670 fr.	16.80 fr.
Q_m = 1.000 t. — —	100	2	717 fr.	9.00 fr.
Q_m = 1.000 t. — —	100	4	670 fr.	8.40 fr.

Table 5 below clearly shews the incontestable superiority of the railway from the point of view of speed of service between the two important business centres considered.

TABLE 5.

Comparative time for a return journey over a distance of 400 km. (250 miles).

METHOD OF TRANSPORT.	Speed.	Time of journey.	Frequency.	Station delays at departure and arrival.	Total time of the single journey.	Minimum time for the return journey.
Railway	360	1 h. 10 m.	0 h. 3 m.	40 m.	1 h. 53 m.	3 h. 45 m.
Flying machine with 4 return journeys.	160	2 h. 30 m.	1 h. 15 m.	40 m.	4 h 25 m	8 h. 50 m.
Flying machine with 2 return journeys.	160	2 h. 30 m.	2 h. 30 m.	40 m.	5 h. 40 m.	11 h. 20 m.
Automobile with 4 return journeys.	80	5 h. 00 m.	1 h. 15 m.	40 m.	6 h. 55 m.	13 h. 50 m.
Automobile with 2 return journeys.	80	5 h. 00 m.	2 h. 30 m.	40 m.	8 h. 10 m.	16 h. 20 m.

VI. — Conclusion.

These several considerations sufficiently underline, we think, the marked value that the aerial railway, when properly adapted to ultra-rapid traffic, has from the point of view of speed, of the total capacity of transport, of the cost, and finally of regularity which is the fundamental condition of the movement of postal matter.

[625 .143.5 (.43)]

Bearing plates with ribs in permanent way work,

By Dr. TH. BUCHHOLZ, Engineer, Berlin-Zehlendorf.

Figs. 1 to 8, pp. 399 to 401.

(*Glasers Annalen.*)

I. — Permanent way with timber sleepers.

The use of ribbed bearing plates in permanent way work constitutes a new system of rail fastening, in which the holding down of the rail on to the bearing plate is quite distinct from the fixing of the bearing plate on the sleeper. This system, in view of its solidity of construction and durability, is particularly suitable for lines which have to carry a large number of trains running at high speeds.

The principal advantages of this new system of rail fastening over those generally employed hitherto are as follows:

The timber sleepers are protected at the rail seats against premature mechanical deterioration and against rotting;

the laying and maintenance of the permanent way are simplified, and therefore become cheaper;

the rail, being gripped between the ribs of the bearing plates, can better withstand the effect of high speed running, and is more secure against acts of « sabotage ».

Let us now, using comparisons as necessary, examine these advantages more closely.

A. — Life of timber sleepers on the old types of permanent way.

Long years of experience of the old types of track have shewn that timber sleepers have a relatively short life on the main lines. This fact is almost always attributable to the deterioration of the sleepers under the rail or under the bearing plate. The causes are to be found in the usual method of fastening the rails and bearing plates to the sleepers, the effect of which is that the movements of the foot of the rail, due to the traffic, directly cause the coach screws to become loose quickly. Further, the coach screws, being fixed quite close to the foot of the rail, are subjected to an abnormal strain through the overturning moment resulting from the alternating lateral thrusts exerted by the vehicles.

The early loosening of the coach screws is furthermore helped by the fact that they have to fulfil two contradictory conditions. On the one hand, in view of the feeble resistance of the wood, they should not be subjected to too severe a strain; on the other hand, they must fasten the rail as firmly as possible to the bearing plate.

The shouldered bearing plates (fig. 1), fitted on track of the German State Railways, was an early experiment to find a remedy for the disadvantages attendant upon this system of rail fastening; but the desired object has been only partially achieved. On this track in the first place there is no intimate contact between the foot of the rail and the shoulder of the bearing plate. As a result to the play which exists, the foot of the rail vibrates under the passage of heavy loads,

and rapidly wears away the shoulder. The same thing happens with the holes for the coach screws which are intended to fasten the rail; water can easily penetrate into the enlarged holes, with the result that the wood round the holes begins to decay. To this must be added the strain to which the coach screws are subjected by reason of the lateral thrust on the rails, as the screws are still fixed quite close to the foot of the rail. The premature and rapid wearing out of the metal parts of the permanent way and of the timber sleepers is the inevitable consequence of the play referred to above.

Various railway administrations have endeavoured to prolong the life of sleepers by re-adzing and dowelling, but this method is not attended with any appreciable success, as it is not possible to ascertain sufficiently soon the state of the sleepers under the bearing plates, and therefore the deterioration and decay are usually already too far advanced. On the other hand, the real cause of the defect cannot be eliminated, for the direct transmission of the movements of the foot of the rail to the coach screws, and the excessive strain on the latter due to the lateral thrust on the rail, remain.

The defects above noted are still less likely to be remedied with the old types of track, now that the conditions of service are becoming more and more severe owing to the continual increase in axle loads and speeds. In fact, in view of the long life of the track axle loads of 25 tons and speeds exceeding 110 km. (68 miles) per hour must in future be considered.

The old types of track more particularly offer insufficient resistance to the hunting movements of the stock, caused by the increase in speed, and to the tendency of the rails to tilt, caused by these movements.

On electrified lines, moreover, the recognised jerking movements on curves produced by the low lying bulk of the motors of the locomotives and motor coaches must be taken into account.

B. — Life of timber sleepers on track fitted with bearing plates having ribs.

With the ribbed bearing plate the premature loosening of the coach screws and plates is prevented by the independent fastening of the plates to the sleepers and of the rails to the plates, so that the movements of the foot of the rail do not react directly on the coach screws fixed in the wood (fig. 2). These movements are absorbed by the special bolts, which do not rest upon the soft wood of the sleepers but fit into slots in the metal bearing plates. The coach screws themselves are clear of the foot of the rail, and are thus preserved not only against direct shock from the rails, but also against excessive strain from the tilting moment of the rails (1).

In this manner, and as further the large supporting surface of the plates offers great resistance to lateral displacement along the sleeper, the coach screws are left free to perform perfectly their real function, which is to fasten the plates firmly to the sleepers and to prevent the plates from moving up and down, and consequently damaging the surface of the sleepers and the sides of the holes.

In the case of curves, where use is made of gauge adjusting keys, the ribbed bearing plates have been given a particularly large seating (fig. 3). This is an advantage as regards the sleeper in view of the greater strain to which it is subjected on curves. The coach screws hold better in track fitted with ribbed plates, as it is possible to drill the sleepers both for straight sections and for curves before they are laid, so that the sides of the holes (which are the most heavily taxed part of the permanent way) are always thor-

(1) Thus, for example, on permanent way which has shouldered bearing plates, the coach screw, which is intended at the same time to hold the rail in position, is placed under tension by a lateral thrust of the vehicles of 20 % of the wheel load, whereas on permanent way with ribbed bearing plates this only amounts to 42 % of this load.

oughly impregnated and therefore effectively protected against rotting.

From what has been said above it follows that when ribbed bearing plates are used, timber sleepers are so well protected against deterioration thet they wear just as well at the rail seats as at any other part. It can therefore be assumed in general that the life of the sleepers will equal that of the rails on main lines. The premature replacement of sleepers, which has to be effected at present on a large scale and which causes frequent interruptions in the service, will in future be limited to exceptional cases. The smaller consumption of sleepers will mean considerable economy, which will be particularly welcome in the case of countries which are obliged to import their sleepers. It is not necessary to add that there is scarcely any country where a reduction in the consumption of timber does not appear desirable.

On track using ribbed bearing plates, the widening of the gauge necessary on curves is very easily obtained on the spot by the insertion of simple metal adjusting keys. The great advantage of this is that all the bearing plates, including those intended for use on curves, can be fixed to the sleepers in advance at the impregnating centres, so that the sleepers arrive at their destination ready for laying [1]. Instead of the present practice of boring the holes on the spot (an operation which takes up much time and is not always accurately performed), and instead of the complicated process of fixing the plates by hand, the work can henceforth be done rapidly and accurately by machine at the impregnating centres, which will mean a considerable saving in wages (fig. 4).

When loading and transporting sleepers to which the bearing plates have previously been fixed, the ribbed plate has the advantage that the external ribs are of the same height as the heads of the coach screws, which are thus protected against damage through knocks.

C. — Laying and maintenance of the track when bearing plates with ribs are used.

The laying of the permanent way with ribbed bearing plates is an extremely simple process. This means a further considerable saving in wages. The rails are lowered between the ribs and immediately assume their exact position, for, in view of the mechanical fixing of the plates at the impregnating centre, nothing is required in the way of gauge adjustment.

The placing into position of the gauge adjustment keys, which are no longer used except at curves of less than 300 metres (15 chains) radius (on the German State Railways, for example, such curves represent less than 1 % of the total length of line), is very easily effected by using tie bars; the same applies to the fixing of the clips and clip bolts, and the tightening of the nuts. One important advantage of the use of ribbed bearing plates is that the heads of the coach screws fit without play in the accurately milled slots of the ribs; the contact surfaces are good, so that with only a moderate amount of attention there can be scarcely any wear. Defective screwing in, or twisting of the coach screws, is impossible.

Similarly, the clips can neither twist nor slip, because the wings are held laterally by the ribs. The surface of contact

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(1) Only two classes of sleepers are used : those for straight sections and those for curves. The latter can, however, also be used on straight stretches, *i. e.* the plates for curves are so arranged that by regulating the gauge it is possible to get down to the normal gauge. The advantage of this is that when preparing the plan for laying curves, there is no need for undue precaution as to providing for a change of sleepers immediately on entering upon a straight section of line. It is therefore advisable, in order to provide for the transition from curve to straight, to supply a few extra curve sleepers.

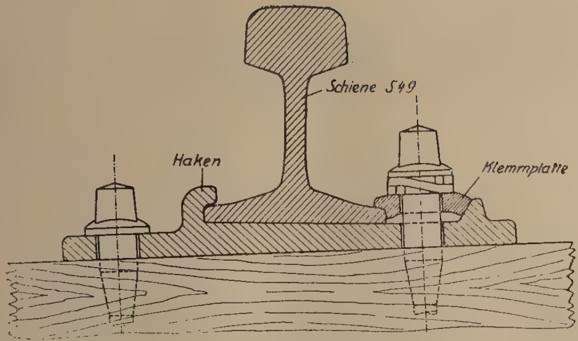

Fig. 1.

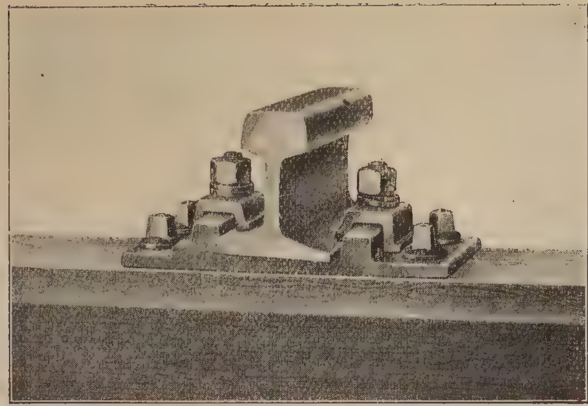

Fig. 2.

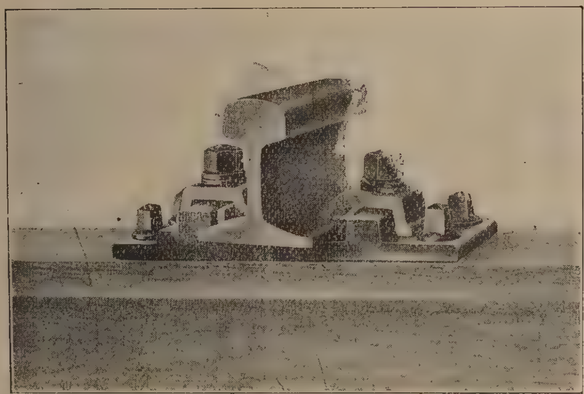

Fig. 3.

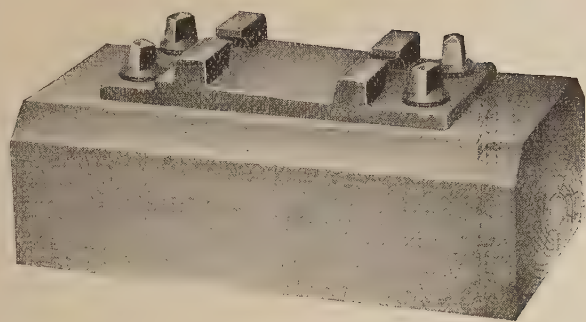

Fig. 4.

Explanation of German terms : Haken = shoulder. — Klemmplatte = clip. — Schiene = rail.

of the clips is of such dimensions that when the coach screws are tightened up, the adhesion between the clips and the foot of the rail or the bearing plate is in itself sufficient to prevent any movement of the rail whether lengthways or sideways.

In cases of relaying of the track or turning the rails when worn on one surface, or when clips or clip bolts are no longer fit for use, the new system has the further very considerable advantage that the work can be done from the side of the track. Up to the present it has been necessary in such cases to detach the bearing plates, with the result that the seating of the bearing plate on the sleeper is less good.

When having to relay lines carrying a heavy traffic, it is to be noted that, it is possible to recommence running without waiting until all the clips are fixed and screwed up owing to the rails being held in position between the ribs.

The insertion of the head of the clip bolt does not cause any weakening of the bearing plate, there being no holes, as is the case with the other types. There can, moreover, be no infiltration of water, causing the sleepers to rot and the clip bolts to rust.

As all the fastenings can be inspected from above, it is easy to verify the assembling of all the parts, particularly the gauge plates (fig. 5). It is unnecessary to add that this facility of inspection is

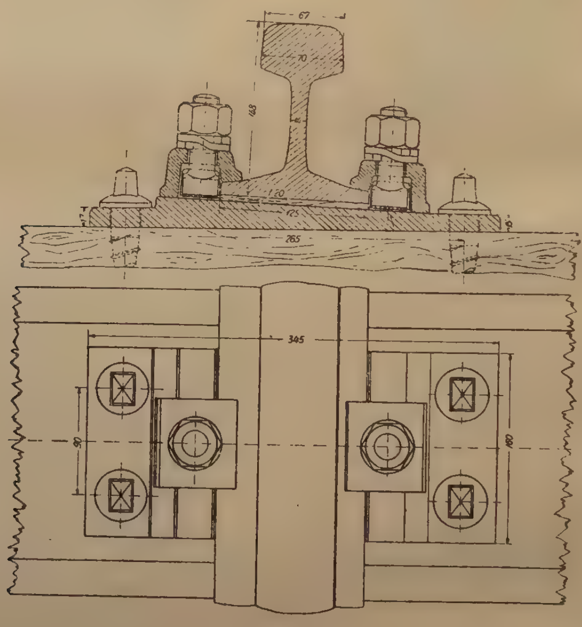

Fig. 5.

a great advantage from the point of view of maintenance. We may remark here that the tightening up of the clip bolt is, owing to the firm, rustproof seat of the boltheads in the slots in the ribs (*i. e.* metal on metal) much less frequently necessary than in the case of coach screws in wood which yields.

A further important point is that the gauge keys, having a greater length of contact against both the ribs and with the foot of the rail — particularly when

made of sufficiently hard metal — are not subjected to wear, as there is no movement : they therefore ensure the permanent tightness of the rail fastenings.

Finally, as regards the question of safety, the track with ribbed bearing plates has the advantage over all the other systems so far adopted. The great length of contact against the strong ribs is a guarantee not only against widening of the gauge through wear, but also against acts of « sabotage », such as have been committed in the Polish « Corridor » and in the vicinity of Leiferde. The ready means by which a train can be criminally derailed by merely displacing a single rail-end, no longer exists when ribbed plates are used.

Ribbed bearing plates are also specially suited for use as double plates at the rail joint, where, owing to the high moment of inertia of the continuous ribs, they act as a bridge, and may therefore be used with advantage to prevent bending and wear at the rail joints. A certain elasticity is at the same time preserved, since the joint plates are specially undercut at the rail-ends. A considerable length of the rail-ends is held between the ribs of the joint plates, offering resistance to lateral movements of the rails and consequently reducing the wear on the fishplates.

One special advantage of the use of ribbed plates is that if elastic packings (pieces of poplar, felt or cloth pads, etc.) are used between the rails and the bearing plates, the ribs of the bearing plates prevent them from slipping outwards (1).

As they are located without play between the ribs, and are held firmly by the clips on either side, the rails with the sleepers form as it were an undeformable frame. There can, therefore, be no creep of the rails, and expensive anti-creep devices are thus unnecessary.

The great rigidity of the frame which forms the track makes it possible to use longer rails, or to join together several lengths of rail, without its being necessary to widen the expansion gap, since the greater part of the tension produced at high temperatures by the expansion of the rails is easily absorbed by the rigid track and the resistance of the ballast.

The fixed position of the rails between the ribs also prevents the track getting out of line.

The German State Railways are at present experimenting with 30-metre (98 ft. 5 in.) rails. It is probable that such lengths of rail will be generally adopted in future. The resulting reduction in the expense incurred on rail joint fittings, in the cost of maintenance of the track and rolling stock, and in the consumption of electric power or coal for traction, will be very considerable.

In cases where for special reasons it is not possible to use long rails, the expansion gaps can be reduced, with a consequent reduction in jolting.

The fact that the horizontal expansion of the rails is largely absorbed by the rigid track obtained by the use of ribbed plates, is also of great importance as re-

(1) It is unnecessary to add that, where ribbed plates are used, the use of elastic packings between the foot of the rail and the bearing plate is not absolutely essential. The German State Railways, however, contemplate the use of these linings in the new type of permanent way, for the following very sufficient reasons :

a) Owing to the undulatory movements of the rails, the elastic packing follows changes in the shape of the foot of the rail, and this helps to diminish the tendency of the sleepers to rock ;

b) As the foot of the rail and the bearing plate are not in direct contact at the rail seat, any small irregularities in the metal surface are counter acted, and consequently the rail can be fitted more intimately to the bearing plate ;

c) There is no contact of metal on metal, and consequently there can be no wear at these points ;

d) Running is smoother.

These advantages more than make up for the slight extra expense incurred for the packings.

gards all types of joint with supporting fish-pates. It is well known that the thermic expansion produces its full effect at the joint. It is therefore necessary at times of great temperature variations to loosen the fish-plates slightly. This, however, reduces their load-bearing capacity, and, consequently, the more the thermic expansion is absorbed by the track, the more tightly can the fish-plates be screwed up without having to allow for variations in temperature.

By virtue of its rigidity, a track fitted with ribbed plates affords very smooth running. For this reason the system is particularly suitable for suburban lines and viaducts, where the transmission of vibration and the running noises inseparable from the present type of permanent way are particularly disagreeable.

In spite of the somewhat higher cost of installation, the use of ribbed plates is also an advantage on secondary, industrial and branch lines, as such lines will be increasingly used in the future by wagons of large capacity and heavy axle loads, the number of which in service increases constantly.

Now that it has been found possible, by the use of ribbed bearing plates, to effect the above-mentioned important improvements in lines laid on timber sleepers, the adoption of these plates is to be recommended strongly for those countries which up to the present have laid the rails directly on the timber sleepers, for the extra cost of the ribbed plates is much more than compensated for by the better preservation of the timber sleeper and the resulting reduction in renewals.

It may further be remarked that if track fitted with ribbed plates is more costly than one in which the rails are laid directly on the timber sleepers, the extra expense is compensated in part by the fact that it is possible to have a narrower rail foot, as the grip of the ribs on the rail seeing that the way the rail is held between the ribs greatly increases the safety as regards turning over. One further very important economic advantage may be mentioned, *viz.* that where ribbed plates are used, it is possible to use the less expensive soft wood sleepers.

Railway companies whose limited resources compel them to observe strict economy may enjoy the greater part of the advantages of the new system by using the ribbed plate on all the joint sleepers, and on only one-half or one-

Fig. 6.

third of the intermediate sleepers, as is already the practice on the Austrian and Serbian railways (fig. 6).

For lines on which the rails are laid vertically, *i. e.* without any inclination, on the sleepers, the system of gauge adjustment adopted by the Hamburg overhead railway is of particular interest. The widening necessary on curves is obtained by means only of two kinds of bearing plates, so that special movable gauge adjusters are rendered unnecessary [1].

This last remark applies also to the shouldered gauge adjusting clips with which the German State Railways are at present experimenting.

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[1] This method of adjusting the gauge will, however, only be used on small lines where there are numerous curves, for it loses the advantage derived from the previous fixing of the plates at the impregnating centres.

D. — Comparison of costs.

We will begin by making a comparison of costs as between, on the one hand, the common type of permanent way with heavy rails weighing 49 kgr. per metre (98.78 lb. per yard), fastened by a clip with two holes (1) on to a shouldered bearing plate with four coach screw holes, designated from now on as :

$$\text{S. C. } 49\,\frac{\text{L} + 22\,\text{B}}{15}\ (^2)$$

and, on the other hand, the type of permanent way which has ribbed bearing plates, designated from now on as :

$$\text{S. N. } 49\,\frac{\text{L} + 22\,\text{B}}{15}\ (^2).$$

The cost of the materials, that is to say, the cost of red pine sleepers and of the fittings (bearing plates, clips, coach crews and bolts, spring washers and cramp irons to prevent creep), amounts, according to tables I and II;

a) For a track fitted with shouldered bearing plates

$$\text{S. C. } 49\,\frac{\text{L} + 22\,\text{B}}{15},$$

to 20 219.27 mk. (or, in round figures, 20 220 mk.) per kilometre (32 540 mk. per mile);

b) for a track fitted with ribbed plates

$$\text{S. N. } 49\,\frac{\text{L} + 22\,\text{B}}{15},$$

to 22 448.91 mk. (or, in round figures, 22 450 mk.) per kilometre (36 130 mk. per mile).

The cost of materials is thus 2 230 mk. per kilometre (3 590 mk. per mile) higher where ribbed plates are used than where shouldered plates are used.

This higher cost is, however, recovered by the quicker and therefore cheaper laying of the track by easier maintenance and greater life (as will be seen from the data below), and other appreciable economies are realised. In order to express this in figures, we take below the annual expenditure as applicable to the two types of permanent way, taking into account the cost of materials the laying of the track, the taking up of the old track when the time comes for renewal, interest on the capital expenditure, and writing off of the permanent way materials, but leaving out the proceeds derived from the sale of old material (3). In the hypotheses the annual expenses are calculated according to the formula generally regarded as applicable :

$$A = \frac{(K + K_1)\ 0.0p}{1.0p^n - 1} + K.\ 0.0p,$$

where K represents the initial capital expenditure, K_1 the expenditure incurred when it becomes necessary to replace the old track, *p* the usual rate of interest, and *n* the life of the track.

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(1) In order that the heavy 49-kgr. (98.78 lb. per yard) rail may be retained, at any rate to a certain extent, Krupps specially make clips with two holes.

(2) The formula S. C. 49 or S. N. 49 $\frac{\text{L} + 22\,\text{B}}{15}$ means that for this track use is made of a rail weighing 49 kgr. to the metre (98.78 lb. per yard), one broad joint sleeper and 22 intermediate timber sleepers, per 15-metre (49 ft. 2 1/2 in.) rail.

(3) For the reasons given above, the life of the ribbed bearing plate is very much longer than that of the shouldered plate, the great majority of the latter being of no further use after the first renewal of sleepers, whereas the ribbed plate on the average lasts twice as long as the sleeper. It does not seem necessary to work out the statistical significance of this feature. but it is certain that this advaatage almost justifies in itself the higher initial expense of the ribbed plates.

COST PER KILOMETRE OF TRACK LAID ON TIMBER SLEEPERS.

TABLE I.

S. C. 49 $\frac{L + 22\,B}{15}$

Track fitted with shouldered bearing plates.

Number per kilometre.	DESCRIPTION.	Weight per article in kilogrames.	Cost.		Cost per kilometre.	
			per ton in Reichsmarks.	per article in Reichsmarks.	Reichsmarks.	Total in Reichsmarks.
133 1/3	Rails S. 49 a.	733.04	133.75	...	13 072.73	...
267	Fish-plates Fl. 1	9.06	165.28	...	399.98	...
534	Fish-bolts	0.891	268.50	...	128.88	...
534	Double spring washers	0.092	...	0.04	21.36	...
	Total for rails and fishing, per kilometre of track					13 622.95
67	Double sleepers, impregnated	...	...	15.73	1 053.81	...
1 467	Sleepers, impregnated.	...	...	7.18	10 533.06	...
2 934	Shouldered bearing plates (four holes).	7.800	189.10	...	4 327.59	...
134	Joint plates	20.582	189.10	...	521.54	...
3 202	Clips (two holes).	0.786	208.00	...	523.49	...
6 404	Coach screws 180 mm. long.	0.644	238.00	...	981.55	...
6 404	Coach screws 150 mm. long.	0.538	238.00	...	819.91	...
6 404	Double spring washers	0.092	...	0.04	256.16	...
201	Bolts for fastening double sleepers	2.92	220.00	...	129.80	...
402	Keys.	0.658	...	0.18	72.36	...
800	Anti-creep clamps	4.22	...	1.25	1 000.00	...
	Total for sleepers and fastenings, per kilometre of track					20 219.27
	Total cost per kilometre of track					33 842.22

COST PER KILOMETRE OF TRACK LAID ON TIMBER SLEEPERS
(Continued.)

TABLE II.

S. N. 49 $\frac{Br + 22 B}{15}$

Track fitted with ribbed bearing plates.

Number per kilometre.	DESCRIPTION.	Weight per article in kilogrames.	Cost		Cost per kilometre.	
			per ton in Reichsmarks.	per article in Reichsmarks.	Reichsmarks.	Total in Reichsmarks.
133 1/3	Rails S. 49 a.	733.04	133.75	...	13 072.73	...
267	Fish-plates Fl. 1	9.06	165.28	...	399.98	...
534	Fish-bolts	0.891	268.50	...	128.88	...
534	Double spring washers	0.092	...	0.04	21.36	..
	Total for rails and fishing, per kilometre of track					13 622.95
67	Double sleepers, impregnated.	...	...	15.73	1 053.81	..
1 467	Sleepers, impregnated.	...	...	7.18	10 533.06	...
2 934	Ribbed bearing plates	9.54	210.00 (1)	...	5 877.98	...
134	Joint plates	23.05	215.00 (1)	...	664.07	...
6 404	Clips.	0.751	223.00	...	1 072.50	...
6 404	Clip bolts	0.579	310.00	...	1 149.45	...
12 808	Coach screws	0.538	238.00	...	1 639.82	...
201	Bolts for fastening double sleepers	2.92	220.00	...	129.80	...
402	Keys.	0.658	...	0.18	72.36	...
6 404	Double spring washers	0.092	...	0.04	256.16	...
	Total for sleepers and fastenings per kilometre of track					22 448.91
	Total cost per kilometre of track					36 071.86

(1) These prices may be expected to decrease.

1\. — *Track fitted with shouldered bearing plates* S. C. 49 $\frac{L + 22B}{15}$.

a) Initial capital expenditure : Marks per kilometre.

1. Cost of materials, in round figures	= 20 220
2. Laying of the track, including drilling of sleepers and fixing of plates on the spot (3.60 mk. per metre) (1)	= 3 600
	K = 23 820

b) Taking up the old track after 16 years' service (1.50 mk. per metre) (1) . . . . K_1 = 1 500

c) Interest $p = 6\ \%$ of the capital.

d) Length of service, n = an average of 16 years.

On the above figures, the annual expenses are :

$$A = \frac{(23\,820 + 1\,500)\ 0.06}{1.06^{16} - 1} + (23\,820 \times 0.06) = 2\,415 \text{ mk. per kilometre (3 886 mk. per mile) per year.}$$

2\. *Track fitted with ribbed bearing plates* S. N. 49 $\frac{L + 22B}{15}$.

We will in the first place calculate by how long the life of a track fitted with ribbed bearing plates must exceed that of the other system, if it is to justify its higher initial expense.

a) Initial capital expenditure : Marks per kilometre.

1. Cost of materials, in round figures	22 450
2. Laying of the track, 2/3rds of 3 600 mk. (2)	2 400
3. Drilling of the sleepers and fixing of the plates at the impregnating centre	310
	K = 25 160

b) Taking up the old track after n years of service, 1.50 mk. per metre. K_1 = 1 500

c) Interest $p = 6\ \%$ of the capital.

d) The length of service, n, must be so determined that the annual expenses are the same for the two types of track.

According to the above calculation, the annual cost of track fitted with shouldered bearing plates, with a service life of 16 years, is :

A = 2 415 mk. per kilometre (3 886 mk. per mile).

For a track fitted with ribbed plates the figures are therefore :

$$2\,415 = \frac{(25\,160 + 1\,500)0.06}{1.06^n - 1} + (25\,160 \times 0.06),$$

from which it follows that the length of service

n = about 17 1/2 years.

In other words, for an equal annual expenditure, the ribbed plate track must last for 17 1/2 years, *i. e.* only 18 months longer than a shouldered plate track under identical conditions. The higher expense is then already covered. In practice, however — supposing the utilisation of the track to be the same — the life of the ribbed plate track may be taken as about 50 % longer, *i. e.* about 24 years. On this basis the expense works out as follows:

(1) These figures are regarded as reasonable for heavy track.

(2) Experience has shewn that the laying of the permanent way with ribbed bearing plates takes about one-third less time.

3. — *Track fitted with ribbed bearing plates* $\left(\text{S. N. } 49 \dfrac{\text{L} + 22\,\text{B}}{15}\right)$ *for a period of service of 24 years.*

a) Initial capital expenditure (as in 2) . . . . $K = 25\,160$
b) Taking up the old track after 24 years' service . . . . . $K_1 = 1\,500$
c) Interest $p = 6\ \%$ of the capital.
d) Period of service, $n = 24$ years.

On the above figures, the annual expenses work out as follows :

$$A = \frac{(25\,160 + 1\,500)\ 0.06}{1.06^{24} - 1} + (25\,160 \times 0.06) = 2\,034 \text{ mk. per kilometre}$$

(3 273 mk. per mile) per annum.

We have seen above that in the case of a track fitted with shouldered bearing plates, the expenses amount to

2 415 marks per kilometre
(3 886 mk. per mile) per annum.

The saving is, therefore:

2 415 — 2 034 = 381 mk. per kilometre (613 mk. per mile per annum, merely on account of the longer period of service and the more rapid laying of a track fitted with ribbed bearing plates (1).

To this must be added the economy in ordinary costs of maintenance of the permanent way. Experience has shewn that these costs, in the case of a track fitted with shouldered plates, amount to about 2 150 mk. per kilometre (3 460 mk. per mile) per annum, made up as follows:

a) ordinary maintenance work, carried out each year : about 200 working days per kilometre;

b) scheduled maintenance work, carried out every third year: about 50 working days per kilometre, in addition to the number under *a*).

The yearly average, therefore, for one kilometre is :

$$\frac{(200 \times 3) + 50}{3} = 217 \text{ working days,}$$

or, with a nine-hour working day, and wages of 1.10 mk. per hour, including cost of tools and overhead charges,

217 × 9 × 1.1 = approx. 2 150 mk.
per kilometre (3 460 mk. per mile)
per annum.

As, in track fitted with ribbed bearing plates, the high degree of rigidity prevents the hammering of the rails on the plates, the necessity for re-tightening the screws is less frequent; further, the sleepers show less tendency to jump on the ballast, and therefore the latter deteriorates less rapidly. Estimating these economies at only 15 % of the maintenance charges mentioned above, there is a net saving of 320 mk. per kilometre (515 mk. per mile) per annum.

If to this be added the saving, calculated above, of

381 mk.

arising from the longer period of service

(1) It may perhaps be objected that, with shouldered bearing plates, the sleepers and their accessories can, at the end of 16 years, be used on secondary lines, whereas. with ribbed bearing plates, the sleepers and accessosories can only be re-used after 24 years. This may be considered an advantage as regards the old material of the shouldered plate track. As against this advantage, however, it is to be noted thaf in the case of a ribbed plate track, the percentage of old material which can be re-used, even after 24 years service, is much higher than in the case of the shouldered plate track.

and the shorter time taken in laying the track, we arrive at a total saving of

381 + 320 = 700 mk. (in round figures), per kilometre (1 125 mk. per mile) per annum [1].

The full value of this saving will be realised when it is remembered that (in approximate figures) of the 90 000 kilometres (56 000 miles) of permanent way on the German State Railways, about 50 000 kilometres (31 000 miles) consist of main lines laid on timber sleepers, so that when the installation of ribbed bearing plates is complete the total saving will be approximately 35 million marks per annum.

This comparison of costs confirms the old rule that a slightly higher initial expenditure is subsequently saved many times over if it results in a continuous saving in current maintenance charges.

E. — Conclusion.

To summarise, the advantages offered by the use of ribbed bearing plates may be stated as follows :

1. Greater economy owing to the longer life of the timber sleepers and fastenings and reduction in the cost of laying and maintenance;

2. greater safety — a matter of great importance in view of the increase in the speed of trains necessary as a result of competition by motor vehicles and aeroplanes;

3. greater security against criminal acts, as it is no longer possible to derail a train merely by the lateral displacement of the rail-ends.

The German State Railways consequently have completely abandoned the shouldered bearing plate, and begun to use the ribbed plate on a large scale. A certain number of other railways, in Sweden and Serbia, for example, as also the overhead railways of Berlin and Hamburg, have adopted the ribbed plate although this invention has only become known quite recently.

Recognising the great importance of this innovation to suppliers of railway material, the more important German rolling-mills acquired the patent rights immediately the invention had been made known. Negotiations are in progress with the same object with the largest rolling-mills outside Germany, so that the material may shortly be available for use in all countries.

II — Tracks laid with iron sleepers.

The principal component part of the permanent way, the rail, has a total life of about 45 years. About half this period is in main line service and the remainder in secondary lines service. From the point of view of efficient maintenance, it is very unfortunate that the second principal component part of the superstructure, *i. e.* the sleeper, has a very much shorter life. The iron sleeper, with which we shall now deal, only lasts for about 30 years, even on the excellently maintained Baden track. (The life of the timber sleeper is, of course, considerably shorter than this.)

As a result of the great difference between the life of rails and that of sleepers, it is necessary, during the second part of the rail's period of service, to renew almost all the sleepers, so that

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(1) By capitalising this annual saving of 700 mk. over a period of 24 years, exactly as we have done for the higher initial outlay, by means of the formula :

$$A = \frac{K.\,0.0p}{1.0p^n - 1},$$

it is found that the capital equivalent of the yearly savings is :

$$K = \frac{A(1.0p^n - 1)}{0.0p} = \frac{700(1.06^{24} - 1)}{0.06}$$

$$= \text{about } 36\,000 \text{ marks per kilometre,}$$

or an average yearly saving of 36 000 : 24 = 1 500 marks per kilometre, *i. e.* a yearly total which itself more than covers the extra expense of fitting the permanent way with ribbed plates.

maintenance work on secondary lines is almost continuous.

From what has been said above it follows that it is of the highest importance to make the total life of the sleeper the same as that of the rail, so that, in what is called the second period of service, *i. e.* on secondary lines, the whole of the superstructure may remain undisturbed until the rails are worn to their limit.

This synchronisation is obtained in the new K type of track with metal sleepers on the German State Railways. In this type, the ribbed bearing plates are welded on to the sleepers. The plates thus protect the sleepers at their most vulnerable spot, *i. e.* at the rail seats, against mechanical wear and breakage; further, the cross section of the sleeper where specially liable to rust is strengthened in such a way as to compensate for this.

Metal sleepers to which ribbed bearing plates have been welded present certain other advantages, to which we will refer in detail:

1. — The fastenings (clips, T-headed bolts, gauge keys) are the same as when ribbed bearing plates are used with timber sleepers. It is unnecessary to emphasize the convenience of standardisation of fastenings, from the point of view of the facility of procuring supplies, of the organisation of the stores and, of efficient maintenance and relaying of the permanent way.

2. — The tested types of elastic packing (poplar wood, woven pads, etc.), placed under the foot of the rail, are gripped by the ribs of the bearing plate, which prevents them from slipping out sideways, an inconvenience which has hitherto been unavoidable, especially at curves. Further, the protection of the linings at the sides makes them wear much longer.

3. — The long lateral grip of the ribs on the rail, and the secure fastening of the rail to the sleepers, give the track great rigidity and stability, thereby reducing the strain on the rail joints, fish plates and ballast. In particular, the tendency of the rails to creep is greatly reduced by this increased rigidity. For this reason, and not less because of the good surface contact for the clips, which cannot turn, the use of cramp irons, to prevent creep is reduced to a minimum, if not completely eliminated. This means a reduction not only in the initial capital cost, but also in the cost of laying and maintenance.

The stability of the track is still further increased by the fact that the sleepers with their plates are about 5 kgr. (11 lb.) heavier.

4. — The cross section of the sleeper generally has a uniform thickness of 9 mm. (3/8 inch). No holes are drilled in the sleeper, and this increases the moments of resistance and inertia of the sleepers and plates at the rail seat by about 30 %. Further, there is no longer the danger of the gradual splitting of the sleeper, starting from splintering at the holes; nor is there any damage to the under side of the sleeper holes, due to the movement of the bolt heads. The outer foot of the clip can no longer sink into the face of the sleeper. The abolition of holes, and the welding on of the ribbed plates, give the sleepers at the rail seat, where the pressure also is more evenly distributed, such a load carrying capacity that, the sleepers will not develop faults at the rail seat, even with the heavier loads which are to be expected in the future. Protection against excessive bending is of course very important as regards the life of the sleepers, for it is an undoubted fact that in the past the premature rusting through of sleepers, precisely at the roof-shaped and vertical bars close to the rail seat has been chiefly due to such continued bending. The roof-shaped bars in particular stretch continually under the weight of the wheels. This produces a certain degree of fatigue and porosity in the metal, and

this in turn facilitates the penetration of rust. Experience has shewn that careful attention to details of this nature plays an important part in the efficient maintenance of the permanent way.

5. — The experiments so far carried out with ribbed plates on timber sleepers show that the clip bolts are held very firmly in the rib slots. Where iron sleepers are used this is most important. With the present drilled sleepers, the heads of the T-headed bolts are below the under surface of the sleeper, so that under the passing of trains they knock continuously against the ballast and thereby become loose much more quickly. Fine dust from the gravel then penetrates continually and is ground between the bolt head and the face of the sleeper, resulting in premature wear at this important spot.

This wear is still further accentuated by the rust which forms as a result of the damp rising from the ballast, as well as by the lateral movements of the foot of the rail which movements shake the bolts either directly or through the clips. On the other hand, the wear the clip bolts cause in the rib slots of the ribbed bearing plates is almost nil, as these plates are made of harder steel and consequently the contact surface of the bolt heads in the slots is less liable to wear, the more so as the bolts themselves are, as in the past, made out of ordinary and therefore softer metal. Further, the contact surface between bolts and slots is about 30 % larger than in the present B type track of the German State Railways. Where ribbed bearing plates are used the clip bolts are held more firmly, their heads engage in the rib slots which are always clean and protected against the infiltration of gravel dust and water; there is therefore practically no possibility of wear or rusting. The bolts are also still more secure, because the movements of the foot of the rail are met by the strong ribs of the plates welded on to the sleepers, whereas hitherto these movements very often affected the bolts through the loose gauge plates on the face of the sleeper.

It is well known that in the B type track it frequently happens that the square of the bolt shaft loses its shape to such an extent, by reason of excessive tightening and rust, that the bolt is no longer proof against twisting, which makes it impossible to tighten it up to the required degree. With ribbed bearing plates it is quite impossible for the bolts to twist, for the whole of the bolt head fits without any play into the slot in the rib.

For the above reasons it will not be necessary in future to tighten or renew the clip bolts so frequently. The wear on the bolts and clips will be so slight that the majority of the parts will be fit for use in the second period of service, *i. e.* on secondary lines.

6. — It has already been found that the ease with which track with ribbed plates can be laid on timber sleepers has resulted in appreciable savings, and the same is the case when the ribbed plates are used with metal sleepers, as the ribbed plate bolt is much easier to fix. It is moreover, impossible to fix it badly, as the workman is unable to place the clip in position unless the bolt is correctly inserted. In view of the mutual interdependence of the bolt and clip, the workman can tighten the bolt with one hand without having to hold it with the other hand by means of a metal grip, as is necessary in the case of B type tracks. Further, there is no need to fix gauge adjusters, for, in accordance with a new regulation of the German State Railways, widening of the gauge is only laid down for curves of less than 300 m. (15 chains) radius, and, of course, such curves do not exist on normal lines. On the other hand, on B tracks adjusting clips are still necessary to support the foot of the rail laterally.

Fig. 7. — Iron sleeper (half length) with ribbed bearing plate welded on.

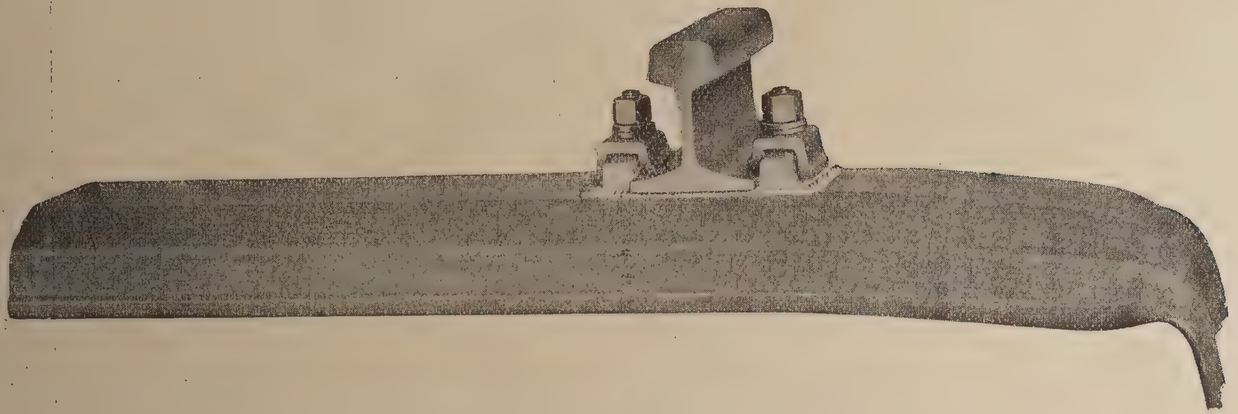

Fig. 8. — Raid laid with ribbed bearing plate on iron sleeper (half length of sleeper).

Comparison of costs.

Below will be found an economic comparison between the iron sleeper of track B and that of track K (designation adopted by the German State Railways for tracks fitted with ribbed bearing plates).

We assume a total period of service of 45 years [1], 22 1/2 of which are on main lines and 22 1/2 on secondary lines.

After the period of service on the main line, the small metal detail parts of the track are renewed as being no longer fit for use, in both types of track; further, in track B, sleepers which are no longer fit for use are replaced by new ones, for example by the lighter Baden-sleeper, which is 75 mm. (2 15/16 inches) deep, and suffices for the requirements of secondary lines.

A. — **Main line.** (Period of service : 22 1/2 years.)

1. — INITIAL CAPITAL OUTLAY

a) *Track B :*

1 sleeper	78.03 kgr. at 134.9 mk. per ton	= 10 53 mk.
4 clips	0.876 kgr. × 4 at 204.9 mk. per ton	= 0.72 —
4 T-headed bolts	0.771 kgr. × 4 at 281.0 mk. per ton	= 0.87 —
4 gauge adjusting plates	0.351 kgr. × 4 at 442.9 mk. per ton	= 0.62 —
	Total. . .	12.74 mk.

b) *Track K :*

1 sleeper	77.81 kgr. at 131.9 mk. per ton	= 10.26 mk.
2 ribbed plates	2.81 kgr × 2 at 280 mk. per ton	= 1.57 —
Welding on sleeper		= 1.40 —
4 clips	0.751 kgr. × 4 at 215.05 mk. per ton	= 0 65 —
4 T-headed bolts.	0.579 kgr. × 4 at 290 mk. per ton	= 0.67 —
	Total. . .	14.55 mk.

The difference in cost in favour of track B is therefore :

$$14.55 - 12.74 = 1.81 \text{ mk.}$$

Capitalised over a period of service of 22 1/2 years at a rate of interest (p) of 5 %, this represents ($1.05^{22.5}$ = approx. 3) :

$$1.81 \text{ mk.} \times 3 = 5.43 \text{ mk.}$$

To this must be added the value, as scrap, of the sleeper and small metal parts of track B, less the scrap value of the small metal parts of track K.

2. — VALUE OF OLD MATERIAL.

a) *Track B :*

1 sleeper. . . 78.03 kgr. — 20 % wear = 62.42 kgr. at 50 mk. per ton.		= 3.12 mk.
4 clips	0.876 kgr. × 4	
4 T-heabed bolts	0.771 kgr. × 4	
4 gauge adjuster plates	0.351 kgr. × 4	
	1.998 kgr. × 4 = 7.99 kgr. — 20 % wear = 6.39 kgr. at 40 mk. per ton.	= 0.26 mk.
	Total. . .	3 38 mk.

(1) This is the average total period of service for rails.

(2) The sleepers of track B, after 22 1/2 years' service, cannot give more than a further 8 years' service, and it is not economic to relay them for this short period.

b) *Track K :*

4 clips	0.751 kgr. × 4 = 3.00 kgr.	
4 clips bolts	0.579 kgr. × 4 = 2.32 kgr.	
	5 32 kgr. — 20 % wear	
	= 4.26 kgr. at 40 mk. per ton	= 0.17 mk.

The proceeds of the sale of old materials show, therefore, in favour of track B, a difference of

$$3.38 - 0.17 = 3.21 \text{ mk.}$$

and the total saving in favour of this track is

$$5.43 + 3.21 = 8.64 \text{ mk.}$$

B — **Secondary line.** (Period of service : 22 1/2 years.)

1. — Initial capital outlay.

a) *Track B :*

1 metal sleeper	54 kgr. at 137.9 mk. (1) per ton.	=	7.55 mk.
4 clips.	0 876 kgr. × 4 at 204.9 mk. per ton	=	0.72 —
4 T-headed bolts.	0.771 kgr. × 4 at 281.0 mk. per ton	=	0.87 —
4 gauge adjuster plates . . .	0.351 kgr. × 4 at 442.9 mk. per ton	=	0.62 —
Average cost of transport (2) of an iron sleeper, 54 kgr. at 6 mk. per ton		=	0.32 —
	Total. . .		10.08 mk.

b) *Track K :*

4 clips as above (A 1*b*).	0.65 mk.
4 T-headed bolts as above (A 1*b*)	0.67 —
Total. . .	1.32 mk.

There is, therefore, a saving in favour of track K of

$$10.08 - 8.64 - 1.32 = 0.12 \text{ mk.},$$

which capitalised at 5 % over 22 1/2 years ($1.05^{22.5} = 3$) gives

$$0.12 \times 3 = 0.36 \text{ mk.}$$

2. — Value of old material.

a) *Track B :*

1 sleeper	54 kgr. — 20 % wear = 43.2 kgr. at 40 mk. per ton. . .	=	1 73 mk.
4 clips	0.876 kgr. × 4		
4 T-headed bolts . . .	0.771 kgr. × 4		
4 linings	0.351 kgr. × 4		
	1.998 kgr. × 4 = 7.99 kgr. — 20 % wear = 6.39 kgr. at 40 mk. per ton	=	0.26 —
	Total. . .		1.99 mk.

(1) The price per ton is increased by 3 mk., because, at 78 kgr. each, there are 12.8 sleepers to the ton, whereas a ton represents 18.5 sleepers each weighing 54 kgr.

(2) The cost of transport must be included, as in the case of track K, there is no renewal, and therefore no cost of transport.

b) *Track K :*

1 sleeper. . 77.81 kgr. -- 40 % wear = 46.69 kgr. at 40 mk. per ton.	=	1.87 mk.
2 ribbed plates 2.81 × 2 = 5.62 kgr. — 40 % wear = 3.37 kgr. at 40 mk. per ton.	=	0.14 —
4 clips, as in A 2*b* . . . . 3.00 kgr.		
4 clip bolts, as in A 2*b* . . 2.32 kgr.		
5.32 kgr. — 20 % wear = 4.26 kgr. at 40 mk. per ton	=	0.17 —
Total . .		2.18 mk.

From the longer period of service there results, therefore, in favour of track K, a total saving of

0.36 + 2.18 — 1.99 = 0.55 mk. per sleeper, or, for 1 kilometre with about 1 600 sleepers,

0.55 × 1 600 = *880 mk. per kilometre* (1 416 mk. per mile).

There are also the economies effected in connection with the laying of the track. On the German State Railways the cost of this work amounts to about 3 000 marks per kilometre (4 800 mk. per mile). Assuming that a saving of only 2 % is effected in the laying of track K, this gives a total saving in favour of track K, over a period of 45 years, of :

$$\left[(3\,000 \times 0.02 \times 9^{(1)}) + (3\,000 \times 0.02 \times 3^{(1)})\right] \frac{1}{1\,600} = \textit{0.45 mk. per sleeper.}$$

The quality of the construction in track K results in a reduction in annual maintenance charges, which amount to about 2 000 marks per kilometre (3 200 mk. per mile); here also, assuming a saving of only 2 %, the total saving over a period of 45 years is :

$$\frac{2\,000 \times 0.02 \times (1.05^{45} - 1)}{1\,600 \times 0.05} = \textit{4 mk. per sleeper.}$$

Taking into account also the fact that the poplar wood packings are better held in place in track K, and assuming that they are renewed once less in each period of service of 22 1/2 years, there is a further saving of

$$[(0.14^{(2)} \times 2) + (0.06^{(3)} \times 2)] \times (4.3 + 2.1)^{(4)} = \textit{2.56 mk. per sleeper.}$$

The full saving with track K is therefore :

From the longer period of service	0.55 mk. per sleeper.
Reduction in laying costs	0.45 —
Reduction in maintenance charges	4.00 -
Lesser cost of poplar wood packings.	2.56 —
	7.56 mk. per sleeper

or 7.56 × 1 600 = (in round figures) *12 000 marks per kilometre* (*19 300 mk. per mile*).

(1) At 5 % interest, the interest factor over 45 years is 9, and over 22 1/2 years, 3 (laying in secondary line track).

(2) Price of a single packing of poplar wood.

(3) Cost of laying and taking out a single packing of poplar wood.

(4) We assume renewal to take place at the end of 15 years; *i. e.* the first period is 15 years, and the interest factor for (45-15) years is 4.3; the second period is 30 years, and the interest factor for (45-30) years is 2.1.

These economies will, however, be increased considerably by the almost complete discontinuance of the use of anti-creep clamps. At present 6 of these are used per rail length, and they cost (the cost of fixing included) $1.50 \times 6 = 9$ mk., or, at compound interest over a service period of 45 years, $9 \times 9 = 81$ mk., or about *3.30 mk. per sleeper.*

It must also be assumed that the cost of the ribbed plates, and of their welding on to the sleepers, will decrease as the methods employed are perfected. It may also be anticipated that the price of the ribbed plates per ton which in the above calculations is taken at 280 mk. will eventually drop to from 240 to 250 mk., or say on the average 245 mk., which will mean a further reduction in initial capital outlay of

$$0.035 \times 2.81 \times 2 = 0.20 \text{ mk. per sleeper.}$$

Further, according to welding experts it may be taken that the cost of welding be reduced from 1.40 mk. to about 1 mk. per sleeper. Adding this to the last-mentioned reduction, the total saving on the plates will be

$$0.20 + 0.40 = 0.60 \text{ mk. per sleeper,}$$

or, over a service period of 45 years, a total saving of

$$0.60 \times 1.05^{45} = 0.60 \times 9 = 5.40 \text{ mk. per sleeper.}$$

It is, therefore, beyond question, even to the most severe critic, that the K type of track has favourable prospects. In the welding shop of the German State Railways ot Wittenberge they have already got down to a period of 3 minutes for welding per plate, whereas in the above calculations the basis taken is 8 minutes per plate.

Siemens-Schukert and the « Allgemeine Elektrizitäts-Gesellschaft » have recently been producing an automatic welding apparatus for welding the ribbed plates to sleepers, which still further reduce the time taken up by the welding operation. The great advantage of this automatic apparatus is, however, that the quality of the welding is not dependent on the work of any individual employee.

Conclusion.

After the introduction of track fitted with ribbed bearing plates fixed on timber sleepers, there was a certain increase in the latter as compared with iron sleepers, and this unfortunately necessitated the importation of large quantities of timber sleepers from Eastern Europe. It is fortunate for the German metallurgical industry that it has now been found possible to combine the advantages of the ribbed bearing plate with the use of iron sleepers. The iron sleeper track will undoubtedly prove ultimately superior to the timber sleeper track from the economic point of view. This superiority will derive principally from the greater durability of the iron sleeper, at all events on lines situated in regions where the atmosphere is free from acids, which is the case as regards from 80 to 85 % of the total railway system. With the use of elastic packings (poplar wood) at the rail seats, running is as smooth on iron sleepers with ribbed plates as on timber sleepers, particularly as there are no fastenings which can work loose.

In view of the great advantages presented by a track in which ribbed bearing plates are welded on to iron sleepers, the higher initial expense is very rapidly compensated for, as is shown by the above comparative figures. Here again is confirmation of the old principle that a good superstructure is the best investment a railway administration can make, particularly when it is remembered that this superstructure has a period of service which last for several decades, and that during this long period unforeseen changes may take place as regards axle loads and train speeds.

[625 .115]

The correction of curves on railways,

By N. GIOVENE,

ENGINEER.

Figs. 1 to 3, pp. 407 and 409.

(Rivista tecnica delle ferrovie italiane.)

1. — The curves on railway lines after a few years use, reveal irregularities which may become extensive and dangerous. These irregularities have their origin either in real deformations, properly so called, occurring after the laying in of the track, or in initial errors in the layout.

In practice, it is necessary to check over and correct the layout of curves. The rules to be followed when doing work of this kind are given in the instructions issued to the permanent way staff. Owing to the different cases that may arise, these rules are sometimes incomplete and require to be elaborated by the Engineers responsible for the maintenance of the line.

It would therefore appear opportune to recall to mind certain general principles and formulæ, together with the method of using them, put forward in recent times (1) to: *a*) accurately measure up an irregular existing curve; *b*) calculate the regular curve that ought to be substituted for it; and *c*) lay out this latter curve on the ground.

2. — The fundamental rule to be followed in future consists in *measuring up and laying down a curve by means of the offsets corresponding to equal and consecutive chords* in opposition to the method of laying out the line most often used when building it (abscissæ and ordinates on the tangent; chords starting from a single point of origin with the corresponding offsets; extended chords).

In an arc of a single circle, these offsets are of equal length; in a connecting curve with increasing radii they increase on the contrary from zero at the tangent until they equal that of the arc of the circle.

Having stated this : if on an existing curve (fig. 1) we measure the offsets f_1, f_2, f_3... of the equal and succeeding chords 0-2, 1-3, 2-4..., and if on a diagram with the chords as abscissæ the offsets are shewn — preferably to a large scale as ordinates, by joining up the ends by a continuous line (fig. 2), it is easy to observe any irregularities indicated by sudden changes, and to establish the lay-out to be substituted so as to get the desired improvement.

The difficulty is this : *to set out the new circle on the site, reference must be made to the existing curve*, and for this it is necessary to calculate the displacements d^1, d^2, d^3... the different pegs 1, 2, 3... have to be given from this last in order to obtain the correct layout, that is to say, the layout which, corresponding to these pegs, in place of the offsets f_1, f_2, f_3 ... gives the new offsets F_1, F_2, F_3 ...

(1) See the article of Mr. Lefort published in the *Bulletin de la Société des Ingénieurs Civils de France* of May 1910 and in the *Revue Générale des chemins de fer* of January 1911. See also the later articles, especially those of Messrs. Cassan, Hallade and Triboulloy, the synopsis published in 1926 by Mr. Chappelet in the review l'*Ingénieur-Constructeur*. See also the article published by *The Railway Gazette* of the 15 September 1916, page 283.

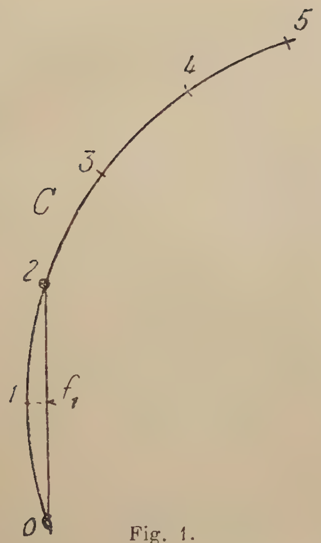

Fig. 1.

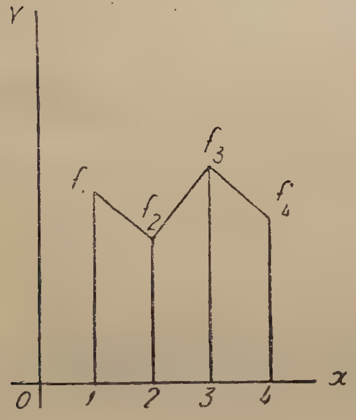

Fig. 2.

If in order to state the difference between the offsets at a point P, we take

$$\delta_p = f_p - F_p$$

the three following fundamental relations remain, it being understood that the offsets corresponding to two succeeding pegs may be considered as being to all intents parallel :

$$\frac{d_p}{2} = \delta_{p-1} + 2\delta_{p-2} + 3\delta_{p-3} + \cdot \cdot$$

$$+ (p-2)\delta_2 + (p-1)\delta_1 \text{. . . . (1)}$$

$$\delta_1 + \delta_2 + \delta_3 + \ldots + \delta_n = 0 \text{ . . . (2)}$$

$$\delta_1 + 2\delta_2 + 3\delta_3 + \ldots + n\delta_n = 0 \text{ . . . (3)}$$

equations also written in the clearer form

$$\frac{d_p}{2} = \sum_{m=1}^{m=p} m\delta_{p-m} \text{ . . . . . (1')}$$

$$\Sigma\delta = 0 \text{ . . . . (2')}$$

$$\sum_{p=1}^{p=n} p\delta_p = 0 \text{ . . . . . . (3')}$$

3. — Formula (1) enables us to calculate the displacement any peg has to make, as a function of the differences δ relative to all the preceding p — 1 pegs.

The other two formulæ (2) and (3) can be combined in a single theorem if, like Mr. Dupuy, we liken the δ to parallel forces in a single plane : *all that is needed is that they form a system in equilibrium in order to connect the new layout with the old.*

(2) and (3) are *closure equations* in this sense that it is essential if they are to be solved, for the two curves, the regular and the irregular, to join together at the ends on the tangent. When used rationally, the two equations have the following meaning :

(4) a) *For the diagrams relating to the curve measured and the corrected curve, the sum of the offsets is the same.*

(5) b) *The areas between the curves of these diagrams and the axis of the abscissæ have their centres of gravity at the same distance from the point of origin.*

It is unnecessary to state that these equations are always approximations, only allowable in so far as the chords are taken as being shorter and shorter as the radius diminishes.

The quantities d will change in sign according as it is a case of movements towards the inside or towards the outside of the curve to be corrected.

4. — In practice, after having drawn

down the diagram of the offsets for the irregular curve, it is necessary to draw on it the line corresponding to the regular curve it is desired to substitute for it. The offsets F are then obtained from it so as to be able to calculate for each point the difference δ between f and F.

In a graph of this kind, a straight line parallel to the axis of the abscissæ corresponds to an arc of circle, and an inclined straight line to a transition curve, the radius of which decreases proportionally from infinity to a constant value.

In order to correct a curve, there is an infinite number of solutions as the two equations (2) and (3) have already shewn us : it is therefore advisable to attach a limiting factor.

The most simple solution, which naturally presents itself to the mind, consists in re-establishing the original circle, connecting it eventually to the tangents by curves of changing radii. But when the offsets measured can form more than one group of values ranging round an average value, it is well to make the new curve with arcs of different circles, provided they can be joined together and to the tangents by transition curves.

Which of the two systems is the one to be adopted in any actual case in practice and, if the second be selected, to what point should the area of circle of different radii be extended cannot be stated in general terms. Only the careful examination of the graph of the existing offsets, drawn to convenient scales, can bring before the mind for the whole the most satisfactory solution; but subsequently only a very detailed calculation, based on the fundamental principles and equations already given, makes it possible either to perfect this solution or after having determined the necessary displacements, to transfer the new curve on to the site on which the irregular curve is found.

The most convenient way of setting out these calculations is substantially the same with the single radius as with the changing radii for the new curve. We will therefore be content to state it briefly for the most simple case of an arc of circle of single radius joined up to the tangents by curves of changing radii.

First of all, in order to calculate the displacements, formula (1) should be used following the very simple method reproduced below :

Number of the pegs.	Difference between the offsets $f - F = \delta$.	Sum of the differences δ (in between the line).	Sum of the figures of column 3. Half displacement (on the line).
1	2	3	4
0	0		
		0	
1	δ_1		0
		δ_1	
2	δ_2		δ_1
		$\delta_1 + \delta_2$	
3	δ_3		$2\delta_1 + \delta_2$
		$\delta_1 + \delta_2 + \delta_3$	
4	δ_4		$3\delta_1 + 2\delta_2 + \delta_3$
		$\delta_1 + \delta_2 + \delta_3 + \delta_4$	
5	δ_5		$4\delta_1 + 3\delta_2 + 2\delta_3 + \delta_4$

According as the displacement is positive or negative it has to be made to the outside or to the inside of the old curve.

The last column of this table supplies the means for calculating straight away the displacement of a peg which it is known *a priori* can be moved but little from its present position : in this way it can be seen at once, without carrying the investigation any further, if the regular curve chosen can be used or not.

If it can, the subsequent procedure is as follows :

A. *Preparatory calculations : a*) Draw up a table in the form given below; *b*) fill in columns 1 and 2 (measurements taken on the ground); *c*) fill in column 3 using for each peg the product of the offset and the order number of the peg; *d*) calculate the sum s of the numbers of column 2; *e*) calculate the sum S of the numbers of column 3; *f*) find the ratio $g = \frac{S}{s}$.

Number of the pegs.	Offsets measured on the ground.	Product of the measured offset and the number of the peg.	Calculated offsets.	Difference between the measured and calculated offsets.		Sum of the differences of column 5 (in between the line).	Sum of the numbers of column 6 (on the line).	Displacements	
				+	—			towards the outside (+)	towards the inside (—)
1	2	3	4	5		6	7	8	
	s =	S =							

B. — *Draw down the diagram of the corrected curve. Determine the centre of gravity of the new diagram* (fig. 3). — All that is required is to mark off from the point O the length $g = \frac{S}{s}$ expressed in multiples of the chord. The construction is proved by the formula (5).

C. — *Draw down the diagram of the circular part* (fig. 3). — The radius employed for the original layout of the line will be retained.

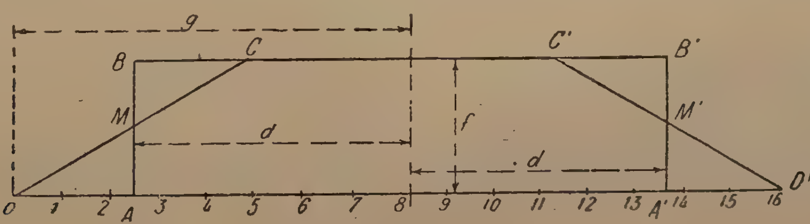

Fig. 3.

After having calculated the offset f corresponding to this radius and to the value chosen for the chord, a line parallel to the axis of the abscissœ will be drawn at a distance f from it. The half-length of the circular part which will connect the two tangents will be, from equation (4), $d = \frac{S}{2f}$ in which S is, as we have said, the total of column 2 and d is expressed as a multiple of the constant chord.

Mark off this length d on both sides of the abscissæ through the centre of gravity. We thus get the rectangle ABB′A′ that is to say, the new diagram having an area equal to the area s of the diagram to be corrected.

D. — *Establish the diagram of the transition connecting curve.* — Take the centres M and M′ of the straight lines AB and A′B′.

Continue OM to meet BB′ in C. Through M′ draw a line M′C′ of equal slope to OC and continue it to D' where it meets the axis of the abscissae.

We get in this way a comprehensive diagram OMCC′M′D′ of the complete corrected curve having an area equal to that of the rectangle, and, consequently, to that of the diagram of the irregular curve measured up, as required by equation (4).

The resulting slope of OMC should comply with the design of the transition curve with varying radii used on the railway in question : if it is not so, it becomes necessary to move the origins O and D′ above the abscissæ axis, that is to say, to borrow still more from the tangents.

E. — *Calculate the new offsets to be inscribed in column 4.* — The calculation which has already been made for the central part of the new curve ought to be extended to the two extreme ends, which can be done graphically, or even analytically, taking into account the slope of the straight lines OMC, D′M′C′ and of their starting or terminating points on the axis of the abscissæ.

F. — *Calculate the displacements.* — *a*) Obtain for each peg the algebraic difference between the measured and the new offsets and enter it in the sub-column of 5 according to its sign. Check if the total of the positive column is equal to that of the negative column; *b*) Enter in column 6 and under the line the algebraic total of all the numbers of column 5 falling on the upper lines as shewn in the sketch given above. In carrying out these operations turn by turn, it will suffice if to the last number found the following number of column 5 is added. The final total should be nil, and any different result shews there has been an error in the calculation; *c*) Use the numbers in column 6 as we have described for those of column 5 and enter the results in column 7. The numbers found are on a line with those corresponding to the pegs; *d*) Double the numbers found in column 7 and enter them in column 8. In this way, we get in amount and in sign the necessary displacements to correct the curve.

5. — All that seems to be required is to have taken great care in all the details of the necessary calculations for the most simple case of correction with a single radius. It will be of use to consult, in addition, the condensed paper of Mr. Chappelet (see the note at the foot of page 406) if one wishes to have at hand a completely worked out numerical example of this case, as also of the more complexe case, for which, however, there are no precise rules but for which it is a question of applying, according to the indications given by a study of the graph of the offsets, a more or less complicated combination of the methods already sufficiently described.

When there are unavoidable obstacles, such as parapets of bridges, lines of posts, signals, etc. there is a maximum

permissible limit of the displacement of some of the pegs, as we mentioned in connection with the first table of paragraph 4.

Whilst they are based on very simple principles, the practical calculations of necessity become very laborious, but a compensation is found in the resulting ability of being able to determine the displacements necessary for the rational correction of existing curves. In any case, this labour contains a lesson for those who may have considered exaggerated the special examination of the connecting curve with specified points during the construction. This examination is on the contrary often opportune when it is a question of joining up two tangents whilst respecting the actual practical conditions which geometrically reduce themselves to a few elementary cases (1).

The two investigations into the connection with specified points during the construction and the correction of curves on railways in operation, have an important point in common in the question of knowing if and when it is advisable to introduce special parabolic connections in polycentric curves. On this subject we refer to the appendix to an article already mentioned as a foot note, in which we also apply to the polycentric curves the figures given in the Instructions published by the Italian State Railways in 1908 for the parabolic connection between circular curves and the tangents.

Finally, it should be unnecessary to insist on the approximate character of the rules given for the correction of curves. It is, however, a question of an approximation which for the usual cases in railway practice, cannot lead to appreciable errors, and which, in all cases, would assist in making estimates in accordance with the two following hypotheses :

a) Two consecutive offsets are taken as parallel;

b) In each case the transition connection is shewn in the graph of the offset by an inclined straight line.

(1) See N. Giovene : *Raccordo con elementi obbligati nei tracciati ferroviari*, Naples, Merano, 1911, and *Engineering News* of 11 May 1916, page 904.

[625 .234 (.73]

Heating and ventilation of passenger cars, (1)

By Edward A. RUSSELL,
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Figs. 1 and 2, p. 415.

(From the *Railway Mechanical Engineer*.)

All exhaust ventilators now in use vary in capacity according to train speed. This variation is admittedly too great as between a car at high speeds and one standing still. It is also a fact that in passenger cars, even at reasonably high

(1) Abstract of a paper read at the annual winter meeting of the American Society of Mechanical Engineers (Railroad Division), held in New York, 5 to 8 December 1927.

speeds, there will be « dead-air » spots, generally about midway between the floor and deck, and at either side of the aisle, in an open-body car. The regulation of the exhaust capacity and the maintenance of a continuous and uniform change of air throughout the car, regardless of train speed or of wind direction and velocity, would result in increased comfort and economy.

The number of ventilators to be applied to a car should be determined by the capacity of the ventilator and the maximum number of occupants to be carried. The Master Car Builders' Association report and recommendations in 1908 showed that 1 000 cubic feet of fresh air should be supplied per hour for each passenger.

The present overheated and stuffy condition often present in cars standing at stations, can best be overcome by an arrangement whereby steam will be automatically shut off and a forced ventilation simultaneously begun whenever cars come to a stop. This could be further developed so that the amount of forced ventilation would be automatically controlled by train speed.

An efficient ventilation and heating system requires sufficient fresh air admitted by proper intakes, uniformly warmed by an easily regulated or automatically operated heating system, thoroughly circulated for the comfort of occupants, and exhausted by properly regulated mechanical means.

The heating of passenger cars has recently received special attention because of the difficulties that have been introduced by the advent of larger locomotives and the hauling of very long passenger trains. The increase of steam pressures necessary for car heating has rendered rubber hose unsatisfactory and uneconomical for steam connections between cars. Where rubber steam hose is used, a maximum of 10 to 12 cars can be satisfactorily heated. In severe weather it is the practice on some roads to remove the steam hose, particularly on head-end cars, at the end of every trip and replace it with new hose, in order to avoid train delays due to burst steam hose.

A most timely editorial appeared in the *Railway Age*, 8 January 1927, entitled « Car heating both difficult and expensive », in which it was stated that « tests have shown that a modern steel passenger car can be heated with about 2.85 lb. of steam per hour per degree difference in internal and external temperatures. This difference may reach 75° or more, and in a 15-car train, therefore, 3 200 lb. of steam per hour are required, or roughly eight per cent of the locomotive boiler capacity... A passenger car is essentially a room on wheels with a large proportion of window area, and exposed on all sides, and passes at high speeds through blizzards and sleet storms, and all kinds of inclement weather. »

At the close of this editorial this very significant statement was made : « The flow of steam through two-inch pipe in train line is restricted in most cases, owing to the use of couplings between cars with only 1 3/8-inch openings. »

In a later editorial (12 March 1927), on « Heating long passenger trains », the *Railway Age* stated that « with a maximum train-line pressure of 130 lb. at the reducing valve, it has been found impossible to heat the rear cars of a 15-car train to anywhere near 70°. Under these conditions, it has been found necessary to have the initial train-line pressure materially higher to supply steam fast enough to be carried back to the rear car of such a train. The utilization of high steam-line pressures presents a number of difficult problems in design, maintenance, and operation that can only be solved by considerable study and experiment ».

These two editorials indicate the problems that have to be met in car heating.

Car heating requirements.

Essentially there are three requirements for ideal passenger car heating: Sufficient volume and pressure of steam supply from locomotive for the adequate heating of every car in train; full-area steam passage through connections between cars, with minimum friction and freedom from leaks; and correct amount, distribution, and regulation of heating surface in each car to heat the car economically and satisfactorily under all conditions.

Locomotive equipment.

For a number of years a 2-inch steam train line has been used under passenger cars, but the source of supply from the locomotive boiler is still restricted to a 1 1/2-inch outlet at the stop valve, with even smaller area for the steam passage through the valve. This restriction necessarily retards the volume flow of steam to meet car-heating demands in severe weather, particularly on long trains. The use of a 2-inch stop valve, preferably of the angle type, constructed so that the valve seat lifts free from the path of the steam, will provide a greater amount of steam for car-heating purposes.

Essentially, then, this area should be maintained in reducing-valve connections in all steam-heat piping on the engine and in all steam-heat connections between the engine and tender. Tests made by one of the large railroads, using a 2-inch stop valve and a 2-inch pipe to reducing valve, showed pressures at the reducing valve to be more than 10 % higher than those where a 1 1/2-inch stop valve and pipe were used with the same boiler pressure. It is important that the full area be obtained at the boiler outlet to the steam-heat line, and that the steam-heat piping on the locomotive be properly protected with covering.

Of more importance than is often recognized is the pressure-reducing valve used in the steam-heat line on locomotives. The length of train and varying outside temperatures determine the pressure and volume of steam required for car-heating purposes. Considering the maximum steam requirements for car-heating, possibly 8 % of the boiler capacity, it is important that the reducing valve should permit economy in steam consumption under other than the maximum requirements. The pressure-reducing valve, therefore, should be capable of adjustment to any desired pressure, and should maintain that pressure without fluctuation. A most important feature in the operation of locomotive pressure-reducing valves is the item of maintenance cost and the delay to the locomotive while making repairs. This has been successfully overcome by the adoption on many roads of a reducing valve so constructed that the cylinder walls and piston (the principal parts which wear or « wire-draw ») may be quickly renewed without removing the valve from the pipe connections.

With present pressure requirements on steam-heat lines, rubber steam hose cannot be successfully used between the engine and tender and at the rear of the tender. Tests conducted in 1920 by the Montreal Air Brake Club showed a pressure loss between the reducing valve and the rear of tender of 13 % with standard 1 1/2-inch steam hose between the engine and tender, and a pressure loss of 5 % with 2-inch metallic connections. Flexible metallic connections for this service should adapt themselves to any motion in service without restricting the steam areas.

With locomotives thus equipped, including all 2-inch piping and connections on tender, a higher pressure is available for car-heating purposes.

Connections between cars.

The necessity for a full 2-inch steam passage through end valves, metallic conduits (used instead of rubber hose,

TABLE I. — **Data compiled from running tests made during the winter of 1926-1927.**

Connections (1) *between cars.*	Outside temperature, in degrees Fahr.	Average speed, in miles per hour.	Pressure at steamheat gage on locomotive.	Average car temperature, in degrees Fahr.	Average drop in pressure head end of first car to rear of: 5th car.	10th car.	16th car.	Average pressure at 15th car, lb. per square inch.
				With 15 steel coaches.				
2-inch metallic	—9	34	113	66	42	69.2	77.3	23.2
1 1/2-inch and hose . .	10	36	129	67.5	63.4	106.8	116.8	2.6
				With 15 steel sleeping cars.				
2-inch metallic	25	33	128	69.5	50.5	88.5	97.9	15.9
1 1/2-inch and hose . .	13	39	147	59	69.2	114.5	125.8	0.0

50 % of ventilators open; from none to 50 % of intakes open, as required.

(1) Two-inch metallic: 2-inch end valves, 2-inch metallic conduits, and 2-inch steam couplers. 1 1/2-inch and hose: 1 1/2-inch end valves, 1 1/2-inch standard steam hose, and 1 1/2-inch couplers.

which is not practical for high pressures), and couplers between cars has been proved by 41 tests.

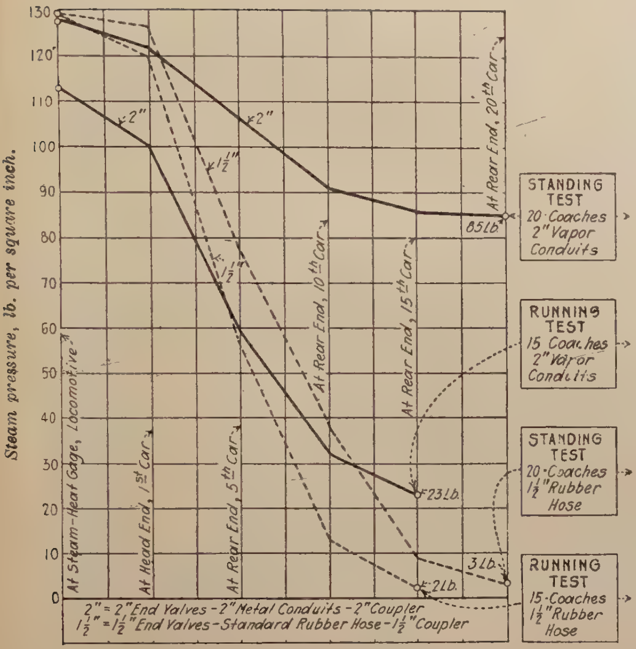
As far back as 1922 it was demonstrated in actual service tests that when changing engines at terminals, steam would pass to the rear of a train equipped with 2-inch connections between cars in less than one-half the time required in a train equipped with 1 1/2-inch connections between cars.

Last winter, one railroad, in an effort to heat long trains of 18 to 20 cars, removed the reducing valve from the locomotive and applied armored steam hose to all cars so that full boiler pressure would be available for car heating; but even with this arrangement the rear cars could not be heated in severe weather. Two-inch end valves and 2-inch metallic conduits were applied, leaving the 1 1/2-inch couplers which have only a 1 3/8-inch gasket opening. However, to heat the rear cars satisfactorily in extreme weather it was found necessary also to apply 2-inch couplers.

Another recent test indicated a pressure drop through the equivalent of five cars equipped with 1 1/2-inch end valves, rubber hose, and couplers which was 50 % greater than with the 2-inch connections between passenger cars.

It has been found, where 2-inch metallic conduits are applied, that the saving in cost of steam-hose renewals over a period of two heating seasons in northern climate will more than cover the entire expense of the metallic conduits. Further maintenance economies will result from the use of metallic conduits constructed so that the gaskets and any other parts requiring renewal can be changed without special tools and without removing the conduits from the cars; and by the use of gaskets with long life—made possible by arranging the conduit so there is no weight or strain on the gasket under any condition of service.

One railroad made a very exhaustive



COMPARISON TABLE.

Average outside temperature, in degrees Fahr.	Average pressure drop per car, in lb.	Time to get steam to rear, in minutes.	Average car temperature, in degrees Fahr.	Average speed, in miles per hour.	Number of deck sash open.	Number of intakes open.
3	2 1/8	4 2/3	69	...	..	...
9	6	5 1/4	67	33.6	1/2	1/2
23	6 1/4	29 1/2	70	...	...	...
13	8 1/3	9 1/2	67.5	35.7	1/2	1/2

Fig. 1. — Variation in steam pressures at different parts of a 20-car train with 2-inch and 1 1/2-inch connections.

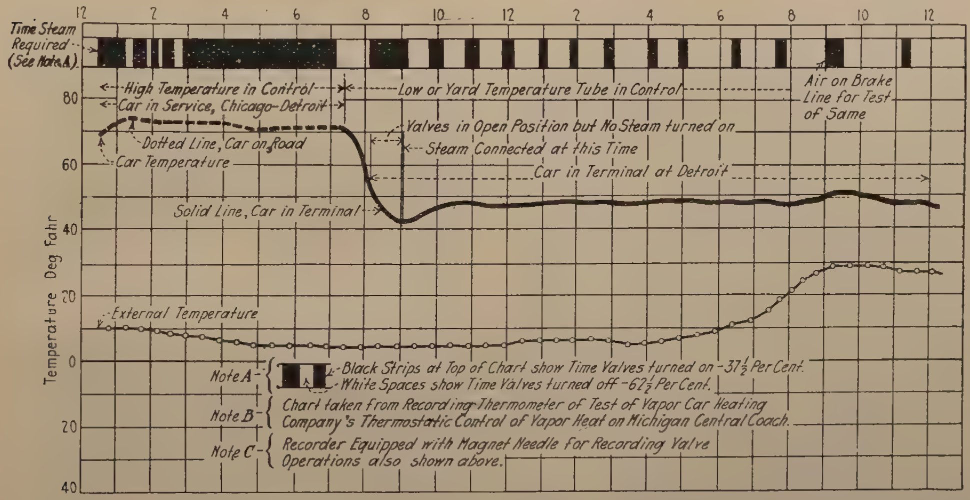


Fig. 2. — Comparison from tests of steam required in service and at terminals by a car equipped with double automatic control vapor system.

(For a car in service, valves are on 90 % of the time; when the car is idle, the valves are on 23 % of the time.)

test last winter, with 20 cars standing in yards and with 15 cars in train service, to compare the results obtained by using 1 1/2-inch connections (end valves, rubber hose, and couplers) and 2-inch metallic connections between cars. Figure 1 and Table 1, compiled from this test, show conclusively the desirability of equipping locomotives and passenger cars with 2-inch steam-heat connections throughout.

Radiating surface in cars.

There is a special field for electric and hot-water heating systems. In strictly suburban service electrically operated cars are heated by electricity, although an electric heating system in more expensive to operate than a steam heating system. For main-line passenger cars, even where hauled by electric locomotives, the use of steam is far more economical, more flexible in regulation, and more desirable.

The use of pressure steam-heating systems has been discontinued. They are neither safe in case of breakage of radiating pipes from any cause, nor economical in steam consumption. With pressure systems the cars in the head ends of trains were always overheated, and the traps were frequently frozen. While steam under pressure has a higher temperature, the difference is of no advantage in comparison to the objections and troubles arising from the use of pressure heating systems.

In calculating the radiating surface required for a pasenger car there are many conditions that must be considered, such as the kind of car, its construction, the tightness of windows and doors, the minimum outside temperatures encountered, and the maximum inside temperature required. There are definite heat losses to be figured. These vary somewhat, depending on methods followed in construction. Formulas for figuring the conductivity of walls, floors, etc., of different materials and insulations, were given in a paper by K. F. Nystrom of the Chicago Milwaukee & St. Paul Railway presented at the February, 1924, meeting of the Canadian Railway Club. This paper also indicated the usual heat losses in steel passenger cars.

The total amount of radiating surface required in a steel passenger car is generally based on the ratio of one square foot of heating surface to 13 cubic feet of space inside the car. This, of course, assumes that the car is properly insulated.

A heating system for passenger cars must be arranged to warm the car under all conditions of outside temperature, which, during a single trip on some roads often range from below zero to a point requiring no heat whatever. This necessitates extreme flexibility in methods of controlling inside car temperatures.

Automatic temperature regulation.

In the development of automatic temperature regulation for passenger-car heating the paramount consideration has been its resulting economy to the railroads. Normaly a passenger car is in train service only one-third of the time. One some roads the average is more, and on some roads less. It is therefore important to avoid all unnecessary waste of steam by the usual overheating of cars lying over at terminals approximating two-thirds of the time.

There is no necessity for maintaining in cars in terminal yards the temperatures required in service. A 50° car temperature in yards will prevent freezing of toilet water and permit cleaning and other work inside the car without discomfort. Such temperatures can be maintained at terminals by automatic regulation.

To maintain by automatic means a temperature of 50° in cars in terminals

for two-thirds of the time, and, say 70° in service for one-third of the time, re quires the use of two thermostats, each made to operate at the predetermined temperature, one for terminal control and one for service control.

The requirements of an automatically regulated vapor system are as follows :

It must retain the « short-circuit » principle of the manually operated vapor system in order to prevent freezing, etc.

It requires a double thermostat properly located in the car.

It must be designed to change automatically from the 50° terminal-control thermostat to the 70° service-control thermostat, or vice versa, without any manual attention This is accomplished by the use of an air-operated contact switch arranged so that the presence of air in the brake train line will put the 70° or service thermostat in control of the heating system, while the absence of air in the brake line will put the 50° or terminal thermostat in control.

It must be arranged so that the 70° thermostat can be put in control manually when it is necessary to heat cars for occupancy and there is no air in the brake line, at the same time not affecting the automatic change from 70 to 50°, or vice versa, when air has again been connected to the brake line.

With the possible maximum use for car heating of 8 % of the total steam generated by the locomotive, it is very essential that every economy be made in heating passenger trains. An automatically regulated vapor system will maintain uniform temperatures in train service, increasing the comfort of the traveling public, and will effect considerable economy in actual steam consumption. It will also eliminate unnecessary waste of steam in heating cars at terminals.

Figure 2 shows the result of a 24-hour test, both in service and at the terminal, of one car equipped with an automatically regulated vapor system. It will be noted that in service, with an average outside temperature of 6°, a uniform temperature was maintained in the car and steam was actually shut off automatically for 10 % of the time. In the terminal, steam was shut off 77 % of the time.

The saving in steam and coal consumed in heating passenger cars at terminals is further emphasized by a test made at a Chicago passenger terminal of two 70-foot steel passenger coaches, identical in construction and in the arrangement and amount of radiating surface, quoted below :

Both cars had arrived in the same train, and inside temperatures were the same at beginning of test.

Ventilators of both cars open; vestibule doors adjusted similarly; temperature readings taken, and all condensation mesured at frequent and regular intervals.

Steam supplied both cars from same source, through a tee-connection.

	Automatic-control car.	Uncontrolled car.
Outside temperature, 26° F.	...	...
Average inside temperature maintained during « lay-over », degrees Fahrenheit	53	83
Percentage of time steam was used in radiating pipes . .	18	100
Average condensation, lb. per hour	33.35	124.76
Average saving condensation, lb. per hour	91.41	...
Or approximately	14 lb. coal per hour.	
Total saving per car per year	40 000 lb. (20 tons) coal.	

Also approximately 37 000 gallons of water.

(Above total saving based on the fact that passenger cars lie over at terminals or in yards practically two-thirds of the time, and allowing 188 days out of each year during which steam would be required in yards.)

Condensation from the yard line drained before steam entered either car.

One car operated continuously under double automatic control vapor system, with the 50° or « lay-over » thermostat in control.

Other car, similarly equipped, had the automatic control feature cut out, and was operating continuously with all valves in the « On » position or wide open, the usual way of heating « lay-over » cars.

In spite of numerous tests that have been made to determine the actual or average steam consumption required to heat passenger cars both in train service and at terminals, it may be said that no formula has been developed that will be accurate under all conditions.

Theoretically, approximately three pounds of steam per car-hour will be required for each degree of temperature difference to be maintained between outside and inside for an ordinary steel passenger car.

The diversity in types of car, their methods of construction and insulation, interior arrangements, number of exposed outlets of heating system, direction and velocity of wind, all affect the condensation of steam for car heating.

An important factor is the speed of train which automatically increases the amount of air exhausted through ventilators. As an indication of the relative increase in condensation with increased speed of train, note the following result of test made :

Outside temperature, degrees Fahr.	Pressure at steam heat gage, lb. per square inch.	Pressure at rear of 11th car : At 40 miles per hour, lb.	At 60 miles per hour, lb.
27	110	40	30
—8	110	30	15

During a test of a train in which all the cars were equipped with automatically regulated vapor system, it was found that as the thermostats in the cars automatically closed the admission valves and cut out steam from the heating coils inside the cars, train-line pressures increased. In one instance, presumably the result of thermostats in all cars shutting off steam about the same time, the pressure at the rear of the last car increased 25 lb. in 10 minutes.

Proper insulation of all exposed piping and connections underneath the car will of course, aid the reduction of condensation in severe weather and at high speeds.

[621 .132.3 (.43) & 621 .134.3 (.43)]

The Schmidt-Henschel high pressure locomotive.

Figs. 1 to 4, pp. 420 to 423.

At the present time endeavours are being made to improve the efficiency of steam power stations by the use of very high pressures. On locomotives this method makes it necessary to depart from the classical type of boiler with interior firebox having flat plates stayed to the outer plates. Investigations of great interest have already been made in this direction, especially in America, where

a test locomotive has been built for the Delaware & Hudson Railroad[1]. Another test locomotive has been built and tested by the German State Railways. We reproduce the details given below of this engine from *The Engineer*.

For the purpose of the experiment it was decided to use a standard type of locomotive, and to make as few changes in it as possible. A locomotive of the S. 10^2 type, No. H. 17206, belonging to the German Federal Railway Company, was selected, and the principal alterations made were the provision of new cylinders and a new two-pressure boiler, in accordance with the patents of the Schmidt'sche Heissdampf Gesellschaft m. b. H. The following are the principal dimensions of the locomotive after conversion :

1. Boiler.

a) *High-pressure boiler.*

Boiler pressure	882 lb. per square inch.
Heating surface of water tubes (fire side)	217.43 square feet.
Diameter of water tubes	1.655 inches inside. 2.01 inches outside.
Heating surface of evaporating coils in high-pressure drum (outside surface)	426.25 square feet.
Diameter of tubes of evaporating coils in high-pressure boiler.	1.26 inches inside. 1.495 inches outside.
Length of high-pressure drum	16 ft. 11 in.
Internal diameter of high-pressure drum	3 feet.
Water content at lowest level	62.15 cubic feet.
Area of water surface.	28.4 square feet.
Height of lowest water level above centre of tank.	3.94 inches.
Heating surface of high-pressure superheater	430.56 square feet.
Diameter of superheater tubes	0.71 inch inside. 0.945 inch outside.
Number of high-pressure superheater elements	30.

b) *Low-pressure boiler.*

Boiler pressure.	206 lb. per square inch.
Diameter of boiler.	5 ft. 3 in.
Length between tube plates.	13 ft. 9 1/2 in.
Number of smoke tubes	116.
Diameter of smoke tubes.	3.02 inches inside. 3.27 inches outside.
Heating surface (gas side).	1 259.38 square feet.
Water content at lowest level.	127.13 cubic feet.
Area of water surface	53.8 square feet.
Heating surface of low pressure superheater.	426.25 square feet.
Diameter of superheater tubes	0.71 inch inside. 0.945 inch outside.
Number of superheater elements.	56.

(1) See *Bulletin of the Railway Congress,* June 1927, p. 552.

c) *Grate area.*

Length	9 feet.
Width	2 ft. 11 9/32 in.
Area	26.59 square feet.

2. Steam engine.

Diameter of driving wheel	6 ft. 6 in.
Diameter of cylinder :	
One high-pressure	11.42 inches.
Two low-pressure	19.7 —
Piston stroke	24.8 —
Volume of cylinders :	
High-pressure	2 271.6 cubic inches.
Low-pressure	7 371.7 cubic inches.
Clearance :	
High-pressure : Front	11.2 %.
Back	11.2 %.
Low-pressure : Front	10.99 %.
Back	10.3 %.
Ratio of cylinder volumes (one high-pressure, two low-pressure)	1 : 6.5.

The high-pressure boiler, which consists of a drum placed above the fire-box, is indirectly heated by internal elements which receive heat from a water-tube fire-box and combustion chamber — see diagram (fig. 1). The fire-box walls

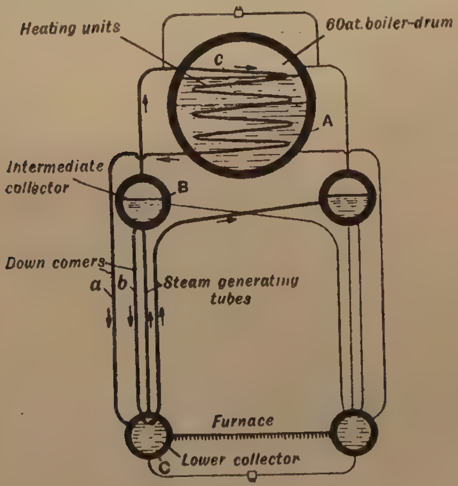


Fig. 1.

consist of tubes, 2-inch external diameter, 1.65-inch internal diameter. They are filled with distilled water, which is constantly in circulation. This water is evaporated in them, and the steam generated passes through heating elements which are situated in the high-pressure drum. The heating steam transmits its latent heat through the heating elements to the water in the high-pressure drum, and the condensate flows back through tubes situated at the back of the tubes forming the fire-box sides and returns to the lower foundation rings. It should be particularly noticed that since the same water is always in circulation, no inscrustation of the tubes can take place, and, assuming that the water is freed of all its contained air, there should be no risk of corrosion.

Intermediate drums are provided between the foundation rings and the heating elements situated in the high-pressure drum to separate the steam and water. The latter, thus separated, passes down tubes which are also situated at the back of the tubes forming the sides of the fire-box. It is thus quite evident that not only is an intensive evaporation

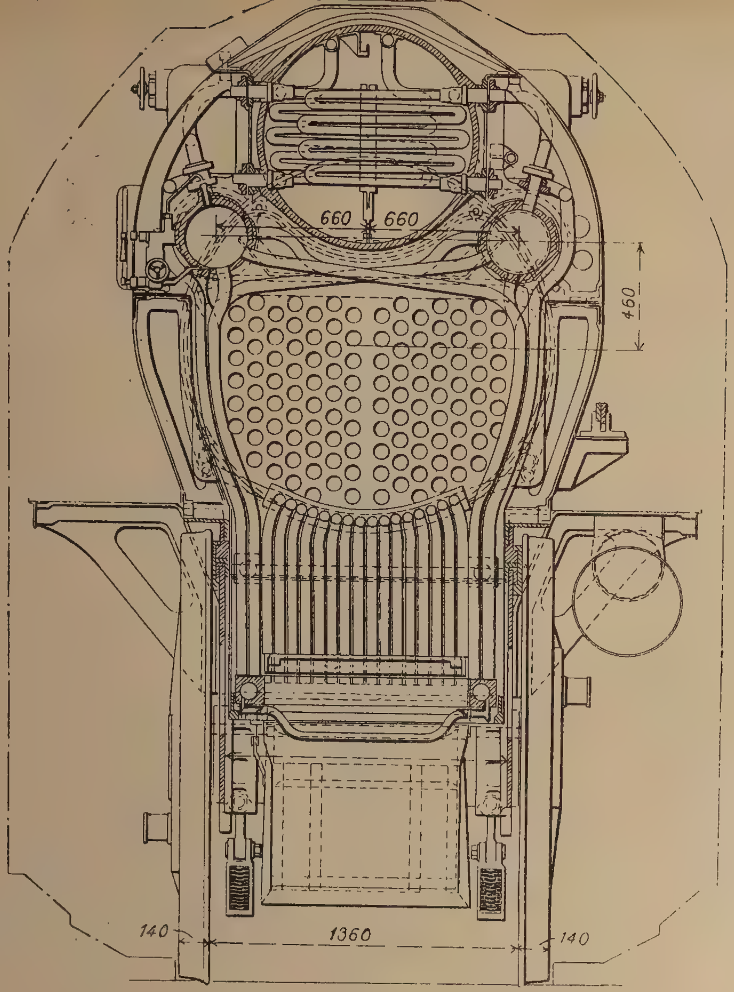


Fig. 2.

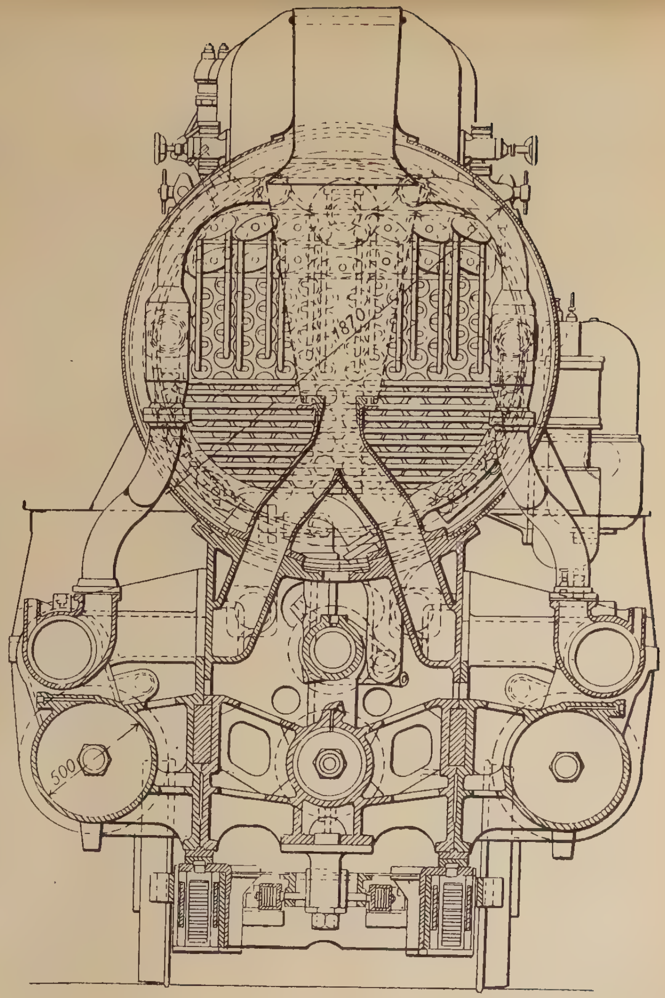


Fig. 3.

Figs. 2 and 3. — High-pressure two-pressure locomotive. — Cross sections.

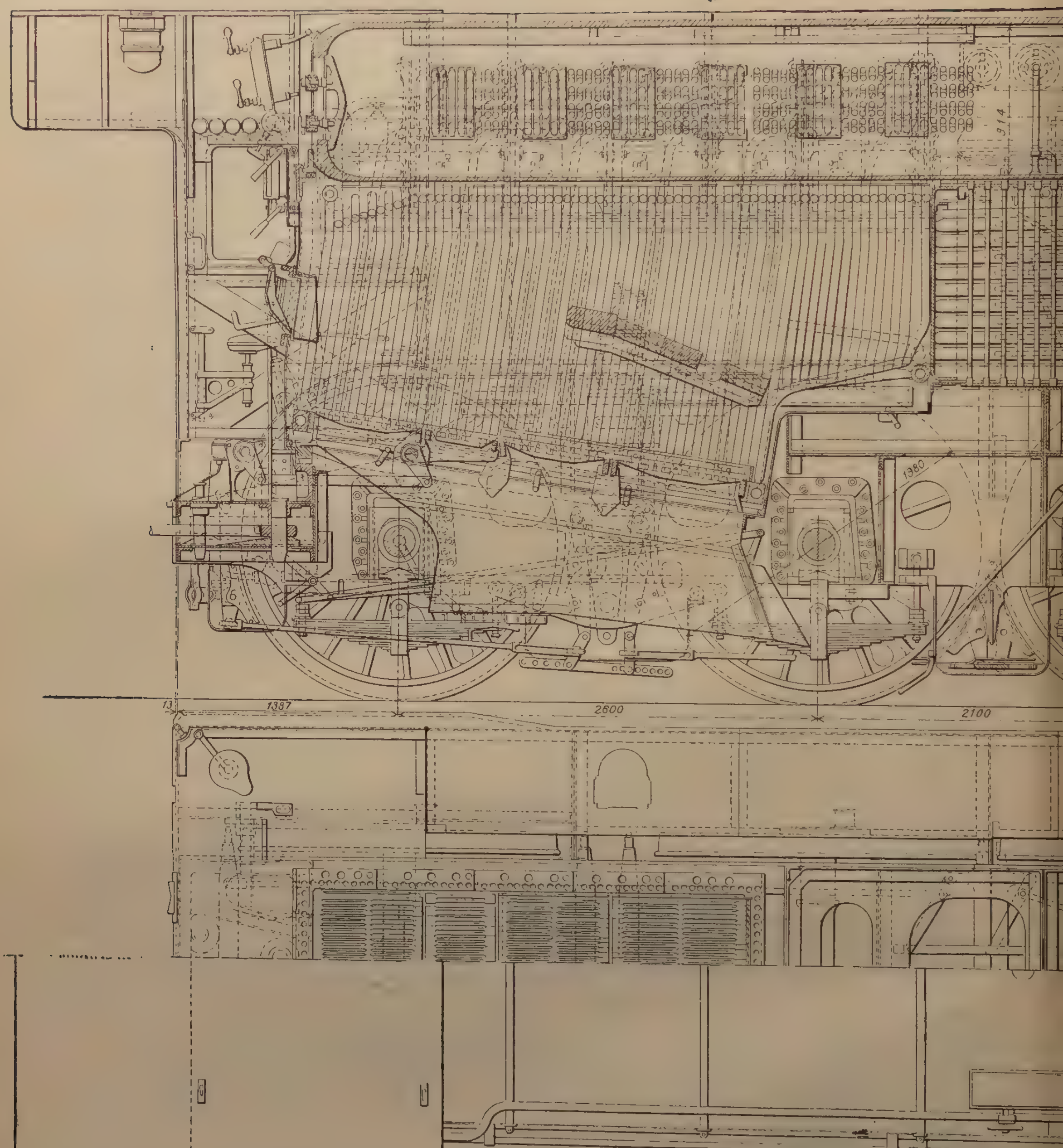
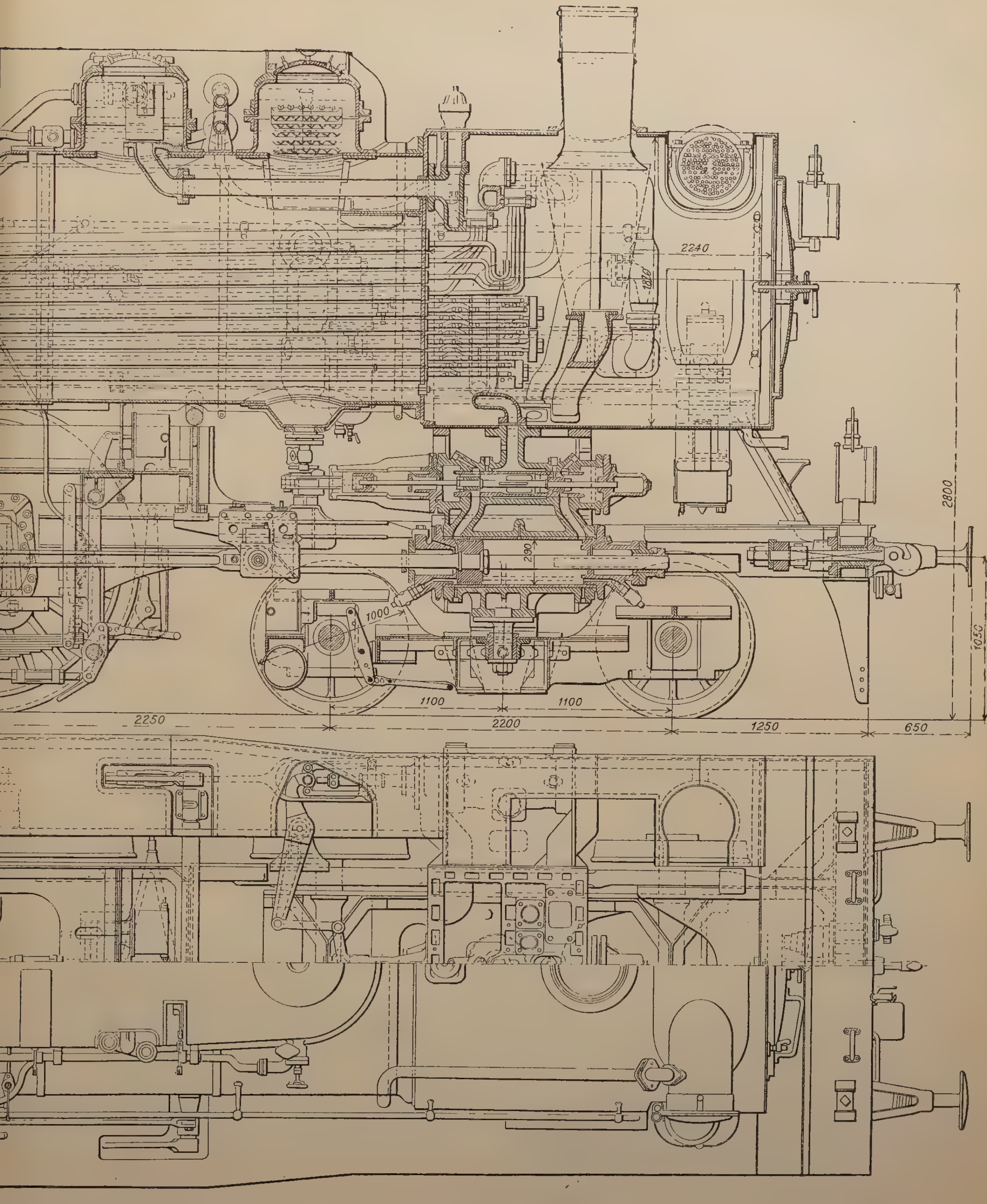


Fig. 4. — Schmidt-Henschel



e two-pressure locomotive.

Tabulated results

Test number.	Date.	Test train. Train number.	Number of axles.	Total weight, tons.	Train section (1).		Average cut-off, per cent.	Total running Distance, miles.	Times, minutes.	Average speed, miles per hour.	Average draw-bar pull, pounds.
1	2	3	4	5	6	7	8	9	10	11	12
1	3.3.27	3 008	47	454	Wp-Mg.		27	69.6	83.2	50.2	5 697
2	5.3.27	Sdpl.	63	640	Wp-Cn.		38	100.7	122.0	49.5	7 822
3	8 3 27	Sdpl.	51	507	Wp-Cn.		30	100.7	121.0	49.9	6 045
4	8.3.27	Pb. 43	51	507	Cn-Wp.		24	100.7	130.0	46.4	4 590
5	10 3 27	Pb. 42	47	463	Wp-Cn.		26	100.7	119.0	50.7	5 247
6	10.3.27	Pb. 43	47	463	Cn-Wp.		20	100 7	131.5	45.9	3 913
7	12.3.27	Pb. 42	67	672	Wp-Cn.		38	100.7	117.0	51.6	7 099
8	12.3.27	Pb. 43	67.	672	Cn-Wp.		32	100.7	120.5	50.1	6 294
9	26.3.27	Pb. 42	67	760	Wp-Cn.		44	100 7	121.5	49.7	9 381

(1) Mg. = Magdeburg. — Wp. =

Tabulated results

Test number.	Average steam pressures, pounds. Water-tube boiler. Section. 1.	2.	3.	4.	5.	6.	Average of 6 sections.	High pressure. Boiler.	Steam chest.	Low pressure. Boiler.	Steam chest.	Blast pipe.	Average inches Smoke-box.	Fire-box.
1	25	26	27	28	29	30	31	32	33	34	35	36	37	38
1	1 029	1 118	1 118	1 279	1 176	1 096	1 135	799	753	150	144	1.62	2.48	1.02
2	956	1 044	1 059	1 279	1 176	1 059	1 096	703	651	154	143	3.68	4.57	1.69
3	971	1 074	1 103	1 309	1 191	1 088	1 122	734	700	151	141	2.21	3.58	0.905
4	985	1 074	1 118	1 309	1 103	1 015	1 100	773	728	151	141	1.47	2.28	0.945
5	956	1 074	1 088	1 294	1 147	1 029	1 097	784	710	151	141	1.62	2.24	0.71
6	985	1 059	1 118	1 324	1 132	1 000	1 103	796	754	151	141	1.03	1.615	0.433
7	953	1 066	1 093	1 368	1 203	1 076	1 126	688	637	150	138	3.38	4.06	1.26
8	1 025	1 129	1 162	1 485	1 306	1 159	1 212	766	728	154	149	2.35	3.23	1.14
9	1 000	1 132	1 132	1 529	1 294	1 074	1 193	681	634	157	146	4.71	4.55	2.127

road tests.

...ne.				Running time under steam.							
...verage draw-bar horse-power.	Mean gradient.	H. P. absorbed owing to gradient.	Total draw-bar H. P. based on the level.	Distance, miles.	Time, minutes.	Average speed, miles per hour.	Average draw-bar pull, pounds.	Average draw-bar horse-power.	Mean gradient.	+ or — H. P. équivalent to resistance on gradient.	Total draw-bar H. P. based on the level.
13	14	15	16	17	18	19	20	21	22	23	24
762.5	1 in 625	6.71	769.2	68.2	80.6	50.8	5 816	787.1	1 in 588	+ 6.90	794.0
031.7	1 in 370	10.95	1 042.6	97.0	114.6	50.8	8 113	1 097.8	1 in 367	+ 11.64	1 109.4
803.9	1 in 370	11.05	814.9	98.0	116.1	50.7	6 217	838.4	1 in 367	+ 11.54	849.9
570.1	1 in 370	10.26	559.8	98.7	125.5	47.3	4 683	592.8	1 in 367	— 10.85	581.9
710.2	1 in 370	11.24	721.4	97.6	112.8	52.0	5 412	749.6	1 in 367	+ 11.84	761.4
480.3	1 in 370	11.15	469.15	98.3	128.0	46.1	4 056	493.2	1 in 367	— 10.55	482.6
979.4	1 in 370	11.44	990.8	96.5	108.2	53.6	7 408	1 057.4	1 in 367	+ 12.23	1 069.6
843.3	1 in 370	11.15	832.1	97.8	116.0	50.6	6 482	876.9	1 in 367	— 11.54	865.4
243.8	1 in 370	11.05	1 254.8	96.6	114.1	50.8	9 777	1 324.7	1 in 367	+ 11.64	1 336.3

...ildpark. — Cn. = Cöthen.

road tests (*continued*)

...cuum, ...ter.		Average temperatures, degrees Fahrenheit.												
		Smoke-box.			High pressure cylinder.		L. P. steam from superheater.	Low pressure cylinder.		Fire-box clothing.		Feed water.		Air temperature.
Aschpan.		Middle.	Rear, top.	Rear, bottom.	Admission.	Exhaust.		Admission.	Exhaust.	Top.	Sides.	Inlet.	Outlet.	
39	40	41	42	43	44	45	46	47	48	49	50	51	52	53
236		574	577	653	680	385	644	514	216	190	189	63	196	55
236		601	651	689	722	417	657	547	223	189	201	59	190	50
158		556	617	649	686	385	622	507	214	187	192	55	201	54
118		545	594	617	649	372	608	482	212	196	196	55	190	54
158		561	617	633	669	374	624	504	210	187	198	55	196	54
078		509	590	612	637	374	601	468	210	192	214	57	187	59
236		610	680	690	716	410	651	540	221	189	212	54	194	50
158		604	653	673	702	392	651	522	216	190	207	55	172	50
133		649	705	720	(748)	457	682	570	244	239	167	57	201	50

Tabulated results

Test number.	Boiler performance.							Steam consumption, pounds.					
								Locomotive.		Pumps.		Locomotive.	
	Total steam, pounds.	H. P. boiler evaporation, pounds.	Per cent H. P. steam to total.	H. P. steam per square foot heating surface per hour, pounds (1) (2).	L. P. boiler evaporation, pounds.	L. P. steam per square foot heating surface per hour, pounds (1) (2).	H. P. steam per square foot evaporating coil surface per hour, pounds (2).	Total steam per draw-bar H. P. hour.	Total steam per draw-bar H. P. hour based on the level.	Air pump.	Feed pumps.	After deduction of steam for pumps.	Per draw-bar H. P. hour after deduction of steam for pumps, based on the level.
1	54	55	56	57	58	59	60	61	62	63	64	65	66
1	19 639	11 332	57.7	37.6	8 307	4.75	19.15	18.6	18.4	310	1 014	18 315	17.2
2	36 250	19 599	54.1	44.4	16 651	6.77	22.6	17.3	17.13	445	1 803	34 002	16.1
3	30 380	16 799	55.3	38.4	13 581	5.34	19.5	18.7	18.45	370	1 532	28 478	17.3
4	25 465	14 396	56.5	30.7	11 069	4.05	15.6	20.6	20.9	390	1 301	23 774	19.5
5	26 664	14 704	55.1	34.2	11 960	4.78	17.4	18.9	18.6	346	1 343	24 975	17.45
6	23 722	14 130	59.6	29.7	9 592	3.47	15.1	22.5	23.0	399	1 252	2 071	21.4
7	33 241	17 813	53.6	42.1	15 428	6.28	21.4	17.5	17.24	390	1 645	31 206	16.2
8	30 825	17 020	55.2	39.1	13 805	5.45	19.9	18.2	18.4	251	1 559	29 015	17.35
9	42 107	21 517	51.0	48.0	20 590	8.07	24.9	16.7	16.6	556	2 026	39 525	15.6

(1) Column 57 calculated on the heating surface of tubes.

(2) These figures are based

maintained in the water tubes, but also a rapid circulation which, in turn, ensures efficient protection of the water tubes, even at the highest firing rates. Half of the water tubes composing the fire-box are arranged in such a way that they cross each other immediately below the high-pressure drum, and thus close the combustion chamber at the top and protect the drum from the influence of the fire-box gases.

The water-tube system of the fire-box was originally divided into six sections, each containing from 17 1/2 tot 24 gallons of water. Subsequent to the trials which were carried out from 2 February to 26 March 1927, by the German Federal Railway Company, these six sections were reduced to two. As a result, there is more uniform pressure throughout, and, consequently, more regular temperature and evaporation, in the high-pressure drum.

The low-pressure boiler is of the ordinary type, having flue tubes, 3-inch diameter, all containing superheater elements. According to actual evaporation trial tests, the high-pressure boiler produced something like 66 % of the total steam at a firing rate of 30 lb. of coal per square foot of grate area. But at a firing rate of 92 lb. per square foot, the corresponding share of the high-pressure steam was only 47 %. The low-pressure boiler also acts as a feed-water heater for the high-pressure boiler. The feed water is drawn from the tender and fed into it in the

road tests (*continued*).

Coal consumption, pounds.			Evaporation per pound of coal, pounds.	Calorific value of coal, British thermal units per pound.	Smoke box gas analysis, per cent.				Excess air, per cent.	(2) Coal per thousand ton-miles—total running, pounds.
Total coal.	Per square foot grate per hour (2).	Per draw-bar H. P. hour, based on the level.			CO_2.	CO.	O_2.	H_2.		
67	68	69	70	71	72	73	74	75	76	77
767	75.1	2.59	7.10	12 760	11.8	0.4	6.6	0.4	46	87.5
291	97.75	2.50	6.85	12 760	11.2	1.2	6.6	0.4	46	82.0
748	69.8	2.28	8.10	12 760	12.2	0.0	7.0	0.0	50	73.5
576	62.0	2.94	7.12	12 760	12.8	0.4	6.0	0.0	40	70.0
230	61.1	2.25	8.25	12 760	12.0	0.2	7.2	0.2	52	69.25
972	51.0	2.88	7.99	12 760	12.4	0.2	7.0	0.1	50	63.75
539	87.5	2.36	7.33	12 760	11.6	0.2	6.6	0.1	46	67.0
343	81.3	2.59	7.10	12 760	12.6	0.2	6.4	0.2	44	64.1
232	116.0	2.46	6 76	12 760	10.8	0.4	7.4	0.1	54	81.5

posed to the fire, viz., 217.43 square feet.
total running times.

usual way. Hence, the scale deposits are, to a large extent, separated in the low-pressure boiler, and there is very little, if any, deposition in the high-pressure drum.

Part of the water which is heated in the low-pressure boiler to the temperature corresponding to the pressure is fed to the high-pressure boiler through a hot water pump. The high-pressure boiler has, in consequence, to transmit little more heat than that required for evaporation, and thus the well-known advantage of a very hot feed are secured.

Should it be required, high-pressure steam can, by means of a valve controllable from the cab, be admitted into the low-pressure boiler. When raising steam, full pressure in the high-pressure boiler can be obtained in 20 minutes. The low-pressure boiler is slower in reaching its full pressure. But if the surplus high-pressure steam be blown into the low-pressure boiler by means of this valve, the working pressure in that boiler can be quickly attained.

The supply of high-pressure steam is controlled by a regulator, which is worked in the usual manner from a lever in the cab. The same regulator rod also controls the supply of steam to another regulator controlling the low-pressure steam. That is to say, the supply of high-pressure and low-pressure steam is controlled by separate regulators, but they are actuated by a single control, common to both.

After leaving the high-pressure regu-

lator, steam passes into a superheater situated in the lower boiler tubes, where its temperature is raised to approximately 750° F. It then enters the high-pressure cylinders. The exhaust from the high-pressure cylinders mixes with low-pressure steam, which has been superheated to a temperature of approximately 800° F. in a superheater situated in the upper boiler tubes. This mixture of exhaust and low-pressure steam is then used in the two low-pressure cylinders.

The operation of this locomotive is similar to that of the ordinary type. There is only one regulator handle, but there are two water gauges to watch and two boiler feed systems to handle. An indirect water gauge is used for the high-pressure boiler, but it has been found that it is not necessary to pay such particular care to the height of the water as in the case of an ordinary locomotive boiler with a crown sheet. In the event of the heating elements becoming temporarily uncovered, no damage is occasioned. The firing, however, is lighter in this method of steam generation, so that there is ample time to attend to these duties.

The valve gears of the high-pressure and low-pressure cylinders are of the conjugated type, the use of which is made possible because the required cut-offs in both the high-pressure and low-pressure cylinders are approximately the same, owing to supplementary low-pressure steam being provided.

The initial trials showed up several defects, but they were of a minor character, and did not affect the design in general. As previously stated, the heating elements were formerly in six sections, and after the trials these six sections were reduced to two. In the table given below, the six sections are numbered separately.

Steam from the low-pressure boiler was found to be relatively wet, and the superheat in the low-pressure cylinders lower than was anticipated. In addition, the full steam pressure in both boilers was not maintained, as the blast pipe action was not satisfactory. Furthermore, the lagging of the water-tube fire-box was not quite air-tight, so that, in consequence of air leaks, combustion was not very efficient. These various defects have now been remedied, and it is anticipated that coal savings will be further increased. Alterations have also been made to the cut-off.

Trials

The trials, of which the results are tabulated above, were carried out by the German Federal Railways, with the aid of a dynamometer car on a section which is practically level. It was unfortunate that during the trials speed had to be reduced over a portion of the section, owing to reconstruction work being carried out. The quantity of feed water was measured by a water meter; the quantity of water pumped from the low-pressure boiler into the high-pressure boiler was also measured. Indicator diagrams were taken every 3.1 miles.

Upper Silesian large coal with a calorific value of 12 760 British thermal units per pound was used. The steam consumption per draw-bar H. P. — see table — is calculated after deducting the steam used for pumps. It must be borne in mind that the speed of this high-pressure locomotive in the trial runs was considerably higher than that of other locomotives, and also that the others were built for heavier duty. The average speed for the running time under steam was 49.7 to 51.6 miles per hour; the actual speed when running unchecked was mostly between 59.9 and 62.1, the difference being accounted for by starting, stopping and slow running owing to the reconstruction work.

The high-pressure feed pumps, as was determined by test, consumed an equivalent of 5.5 % of the water pumped, and the low-pressure feed pump 2 %.

The effective heating surface of both high and low-pressure boilers is calculated on the surface in contact with the fire gases. A moisture of 10 % in the high-pressure drum was allowed for, and it was also assumed that the low-pressure steam reached the superheater with 10 % moisture.

On the constructional and operating side of this design, it should be noted that there are no stayed surfaces, either in the indirect system or in the high-pressure steam drum; consequently, the factor of failure is lessened and the factor of safety increased. The closed circuit system does not require much water, and even if a failure took place in any of the tubes, the result would not be serious, certainly not half so serious as the bursting or blowing out of an ordinary flue tube in the standard type locomotive boiler.

The clearing of any sludge which may be deposited in the drum is quite easily effected, and, as stated above, the indirect heating system is a closed circuit and uses water which is scale and corrosion-free. In practice, the firing up of a boiler with the indirect heating system can be done much more rapidly and with less stressing of the boiler material than with the usual type of steam generator.

As compared with a boiler of the ordinary type for a locomotive, there is little or no increase in the weight of the two-pressure high-pressure boiler. We are told that designs have been got out which show that it can be readily fitted, within the limits of gauge and axle load, to existing types of locomotives in this country. We are also informed that the engine, when stripped after service, showed no greater wear on the various parts — piston-rods, packings, bushings, cylinder walls, piston valves — than would occur in a locomotive under ordinary pressures. Lubrication has given no trouble.

It must be borne in mind that this boiler and the alterations to the engine on which it is fitted were carried out entirely from theoretical calculations. There was no previous practical knowledge of exactly what would be the proper proportions, either of cylinder volumes or heating surfaces required to produce the results which have been obtained, nor was there any practical experience of how the engine would work in actual service until it was built. Some changes — as, for example, the reduction of the groups of high-pressure boiler heating elements to two in place of six — have already been made, and others are in contemplation.

A careful examination of the long table of trial results which we are enabled to give through the courtesy of Dr. Wagner and of The Superheater Company, Ltd., of 195, Strand, London, who handle the design in this country, will well repay the time spent by those who are interested in the details of locomotive performance.

[621 .132 8 (42)]

2 000 H. P. Ljungström turbine locomotive.

Figs. 1 to 17, pp. 431 to 438.

The English edition of the Bulletin for March 1923 gave a description of the Ljungström turbine locomotive built for the Swedish State Railways. A new engine designed according to the same general principles has recently been

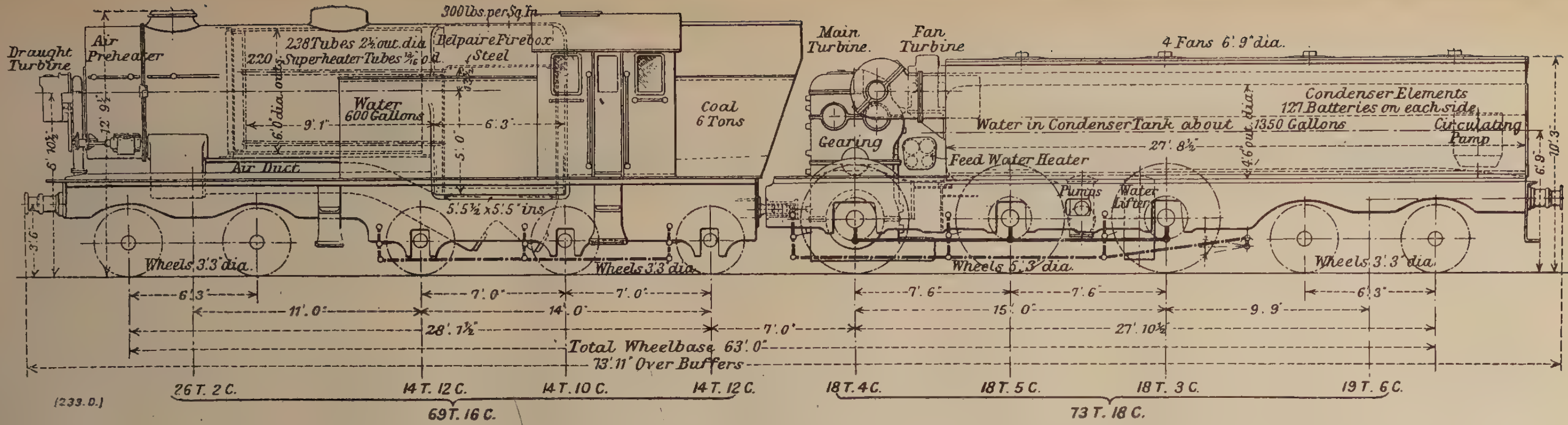
built in England by Messrs. Beyer, Peacock & Co. of the Gorton Foundry, Manchester. This engine differs from the previous one by the fact that certain essential parts have been simplified and improved, and also by the fact that certain safety devices fitted to the first engine have been left off. The following details have been taken from *Engineering.*

The engine was designed so as to be capable of hauling English express trains at a maximum speed of 75 miles per hour, but in actual service, speeds of 76 miles per hour have already been exceeded, and speeds of 80 miles per hour are quite within its powers. In order to gain experience with the new type, the builders made arrangements by which for some months past the engine has been running in regular service on the Midland section of the London Midland & Scottish Railway. As shown in figure 1, its length over buffers is 73 ft. 11 in., and in height and width it is up to the very limit of the Midland loading gauge. When carrying its full load of fuel (6 tons) and 1 950 gallons of water, the engine has a total weight of 143 tons 14 cwt., of which over 54 tons are available for adhesion. Taking the coefficient of adhesion as one-third, it is thus capable of developing a pull of over 18 tons at the drawbar. In service, a drawbar pull of 9.9 tons was recorded during a run made in May 1927 from Derby to Bedford. The load on this occasion consisted of 13 double bogie carriages, the total weight behind the engine being over 400 tons. The engine has also been used on the express service between Manchester and Derby, for which it has hitherto been necessary to use a pilot engine to take the train up the heavy grades between Manchester and Peak Forest. The turbine locomotive proved not only capable of making this run without assistance, but on one occasion, when the train was delayed in starting, made up seven minutes before Derby.

In spite of its exceptional haulage powers, it consumes less fuel than the standard express engine. The Ljungström engine is, moreover, capable of running very long distances without replenishment of its water supplies. One of these engines now in service on the metre-gauge Argentine State Railways has made a continuous trip of over 500 km. (310 miles) without re-watering. In England this property is probably somewhat less valuable than elsewhere, since most of the main lines are now provided with track troughs. Even here, however, the fact that the turbine locomotive can make long runs without re-watering has some very substantial advantages, since it makes practicable a choice of feed waters, and may thus materially reduce boiler-cleaning.

Another valuable characteristic of the turbine locomotive is its large starting torque. The starting torque of a turbine is, in fact, just double the torque developed at maximum efficiency. The above figure can, of course, be increased to almost any extent by the use of a by-pass, short circuiting the high pressure stages. Moreover, there is no dead centre, and the necessity for backing a train in order to secure a start forward, which is today a common experience at some of our railway stations, never arises with a turbine engine. As a further point in favour of the type, it may be noted that recent investigations have definitely demonstrated that impact loads on bridges are predominantly due to the imperfect balance of the ordinary two-cylinder locomotive. On most railways, therefore, there are certain bridges, over which main-line express locomotives are only allowed to run at reduced speeds. The balance of the turbine locomotive is perfect, and it is accordingly unnecessary to subject it to the same restrictions.

It will be clear from figure 1, that the



Total weight in working order : 143 tons 14 cwt.

Fig. 1. — 2 000 H. P. Ljungström turbine locomotive, constructed by Messrs. Beyer, Peacock & Co., Limited, Engineers, Manchester.

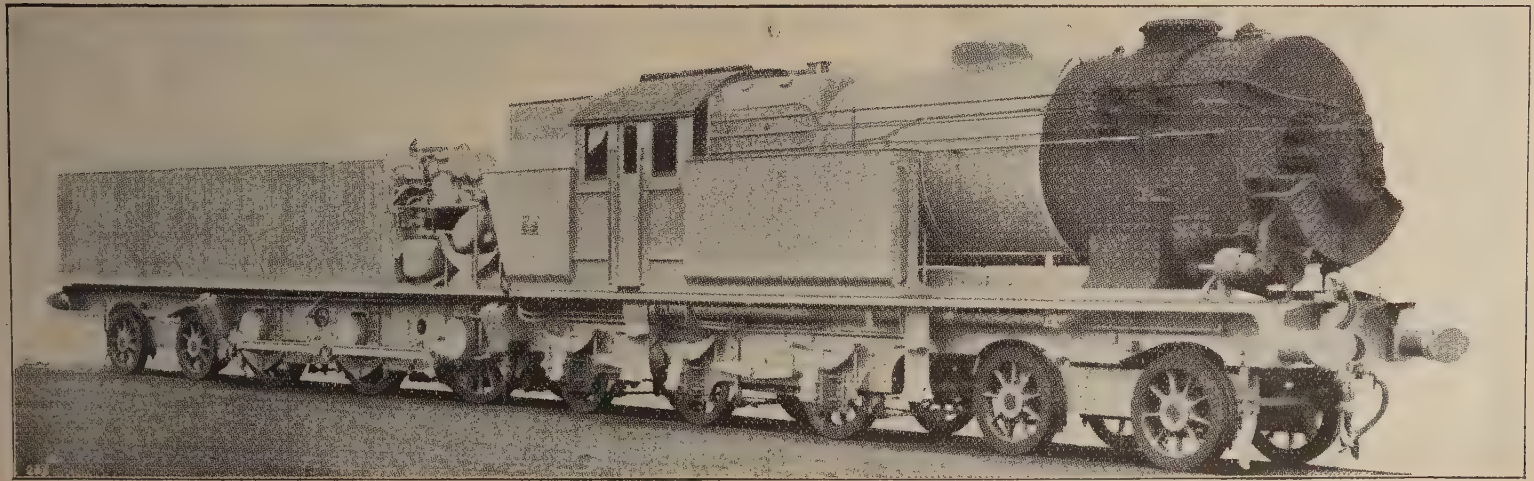
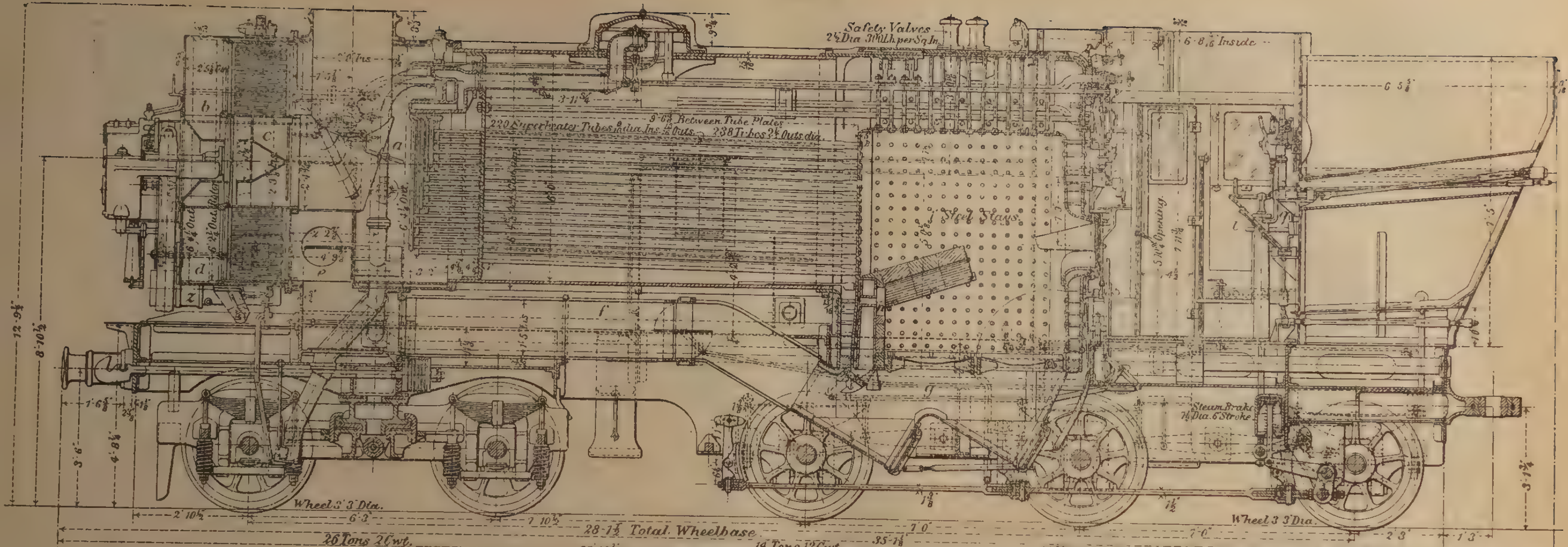


Fig. 2. — General view of engine, complete and in working order.

Fig. 3.


Safety Valves
$2\frac{1}{2}$ Dia. 310 lb. per Sq. In.
6' 8$\frac{1}{16}$ Inside
220 Superheater Tubes $\frac{9}{16}$ dia. Ins. $\frac{7}{8}$ Outs.
238 Tubes $2\frac{1}{4}$ Outs. dia.
9' 6$\frac{3}{4}$ Between Tube Plates
Steam Brake
$7\frac{1}{2}$ Dia. 6 Stroke
Wheel 3' 3 Dia.
28' 1$\frac{1}{2}$ Total Wheelbase
26 Tons 2 Cwt.
14 Tons 12 Cwt.
14 Tons 10 Cwt.
37' 10$\frac{1}{2}$
35' 1$\frac{1}{8}$
12' 9$\frac{1}{4}$
8' 10$\frac{1}{2}$

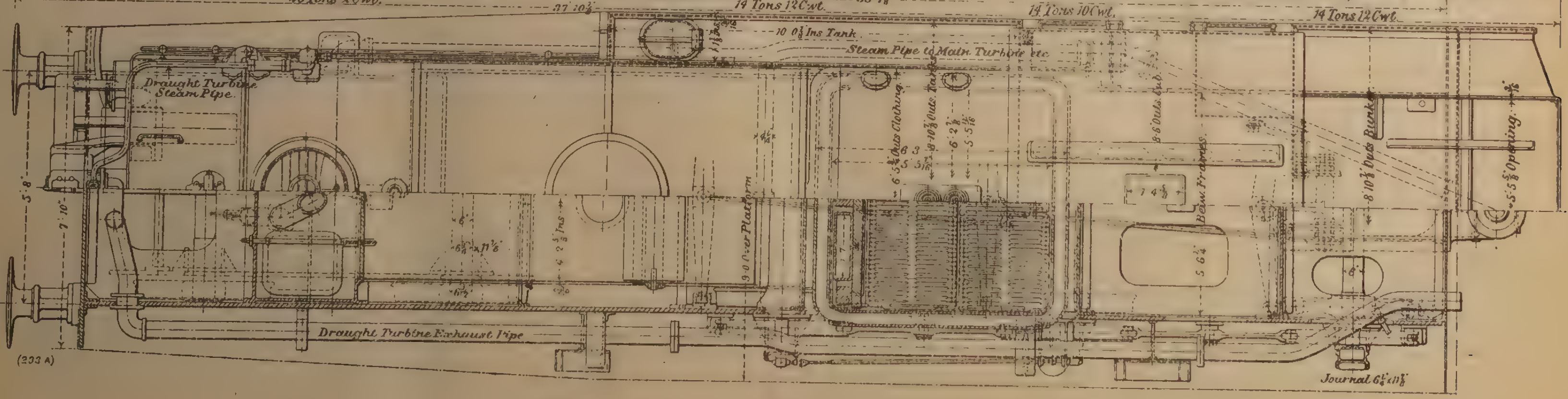
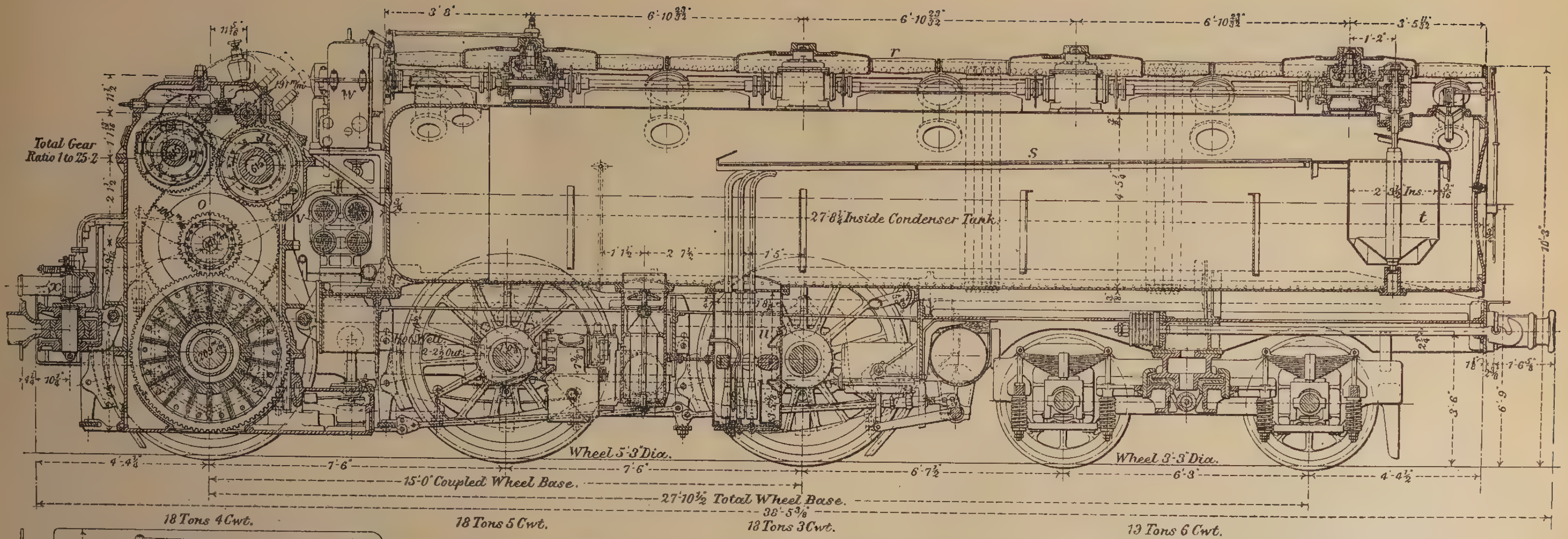
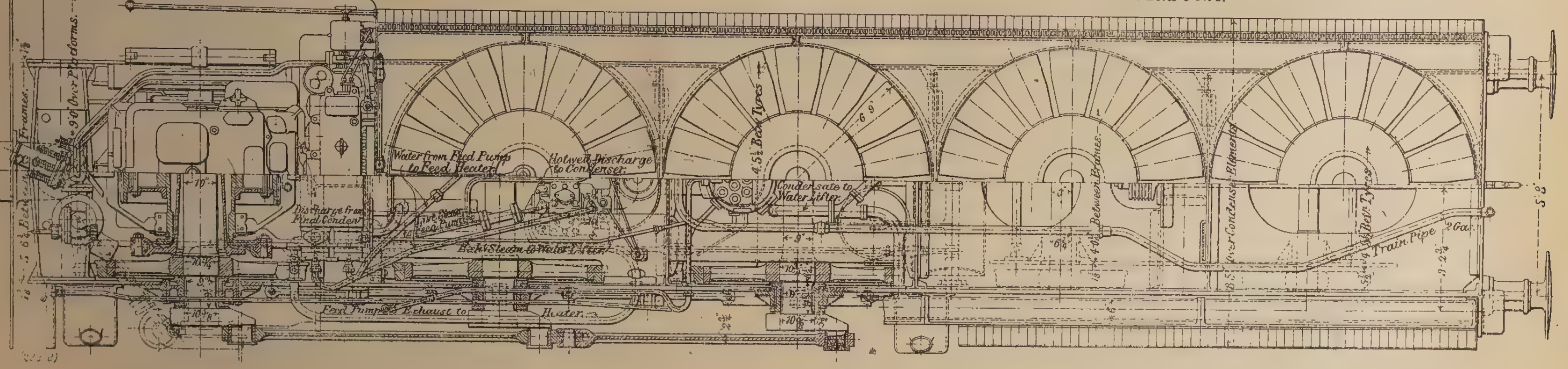

10' 0$\frac{3}{4}$ Ins Tank
Steam Pipe to Main Turbine etc.
Draught Turbine Steam Pipe
Draught Turbine Exhaust Pipe
9' 0 Over Platform
Betw. Frames
8' 10$\frac{3}{4}$ Outs. Bunker
Journal $6\frac{1}{4}$ x $11\frac{1}{8}$
(203 A)

Fig. 4.

Fig. 5.



Total Gear
Ratio 1 to 25·2
27·8¼ Inside Condenser Tank
Wheel 5·3 Dia.
Wheel 3·3 Dia.
15·0 Coupled Wheel Base.
27·10½ Total Wheel Base.
18 Tons 4 Cwt.
18 Tons 5 Cwt.
18 Tons 3 Cwt.
19 Tons 6 Cwt.



9·0 Over Platforms.
Water from Feed Pump to Feed Heater
Hotwell Discharge to Condenser
4·5½ Bent Tyres
Condensate to Water Lifter
Between Frames
Train Pipe
Feed Pump Exhaust to Heater

Fig. 6.

complete locomotive consists of two distinct sections of nearly equal wheelbase, coupled together. The leading section constitutes the boiler plant, whilst the turbine, the reduction gearing, and the condenser are borne by the trailing section, which is mounted on three driving axles and a four-wheeled trailing bogie.

A longitudinal section through the steam generating element is reproduced in figure 3, and half-plans, one being in section, in figure 4. The boiler is of the Belpaire type, and is designed for a working pressure of 300 lb. per square inch. It is 9 ft. 6 5/8 in. long between tube plates. The tubes are of steel, and are 2 1/2 inches in external diameter, with a total heating surface of 1 480 square feet. There is, in addition, 140 square feet of heating surface in the firebox. The superheating surface provided is 640 square feet. The superheater is built up of a series of tubes 13/16 inch in outer diameter. Each element is formed by twice doubling back on itself one of these tubes, so as to form two U bends, one of which is longer than the other. Each of the ducts thus constituted is packed inside one of the boiler tubes, and the steam to be superheated passes first down one leg of the longer U bend and up the other, and then traverses similarly the other U bend, from which it is delivered to a header connected to the main steam pipe. In practice a superheat of 150° F. is easily maintained.

Owing to the obstruction caused by the superheater elements, its is impossible to clean the boiler tubes by brushing, and the operation is accordingly effected by jets of high-pressure steam delivered from a Parry soot blower.

Since the steam is condensed, the necessary chimney draught must be provided otherwise than by a blast pipe. A turbine-driven fan is accordingly installed on the front of the smoke-box, which is extended ahead of the chimney, as shown, so as to provide room for the air preheater. A photograph of the draught turbine and its fan is reproduced in figure 7. The normal speed of the turbine is 28 000 to 30 000 revolutions per minute, and the fan is driven through reduction gearing at 6 000 revolutions per minute. The air is drawn in through a set of louvres arranged in front of the smoke-box. These louvres are interlocked, in a simple but effective way, with the firing door, so that the latter cannot be opened unless the louvres are completely closed. The air which enters by these louvres passes first through a preheater of the standard Ljungström type. This device consists essentially of a drum divided by radial partitions into a series of chambers, which are packed with thin steel plates, ribbed and studded so as to present a very large surface to gases passing through the chamber. The drum is interpolated between two cylindrical chambers, each of which is divided in two by a diametral partition. The incoming air enters the apparatus through one of the semi-circular compartments formed as above described, and passes through the drum into the corresponding chamber on the other side. The hot gases pass similarly through the other half of the apparatus. The drum is kept in constant slow rotation, so that the steel plate packing having picked up heat from the flue gases, is carried round, so as to be traversed by the incoming air, which is thus preheated before it passes on to the furnace. Short circuiting, which might occur, as a compartment of the drum passes the diametral partitions above mentioned, is prevented by guard plates in the shape of sectors, which are sufficiently wide to cover completely one of the chambers of the rotating drum.

Referring to figure 3, the flue gases discharged at *a*, pass round the base of the chimney and through the upper half of the drum shown, where their temperature is reduced to about 150° C. The gases are delivered from the drum into

the chamber *b* and thence are forced by the fan through the central opening *c*, to the uptake. The fresh air is drawn through the louvres into the chamber *d*. It passes then through the lower half of the rotating drum into the chamber *e*, and thence by the duct *f*, to the closed ashpit *g*. A by-pass is provided by which about 25 % of the incoming air is diverted directly into the fire-box through the port *h*. The smoke-box extension, in which the preheater is housed, is mounted on hinges, so that access is easily obtained to the drum of the preheater and to the fan. The drum rests on 6-inch rollers, which are coupled by chain gearing to the fan shaft, and these rollers drive the drum by friction at about 6 revolutions per minute. For use when raising steam or when cleaning fires, a ring blower is fitted at the base of the chimney.

The fire-box plating is of steel 7/16 inch thick. The grate area provided is 30 square feet. In addition to the brick arch shown in figure 3, four large water tubes are fitted inside the fire-box.

The regulator shown in the steam dome is used solely for emergency purposes, the engine being controlled by an independent regulator mounted on the trailing section of the locomotive, but operated from the footplate by the handle shown at *i*. Just above this can be seen the handle *q*, by which the reversing gear is operated. Both controls are within easy reach of the driver's seat. Independent control valves are provided for the blower turbine, the condenser fan turbine, and for the feed pumps.

The bunker holds 6 tons of coal, and is arranged behind the foot plate. The two side tanks, best seen in the general view, figure 2, hold 600 gallons of water and about 1 350 gallons more are contained in the condenser tank, to be subsequently described. The brakes for the six trailing wheels of the boiler carrier are steam operated. The engine is also equipped with standard vacuum brake fittings, the discharge from the air ejectors being passed into the feed heaters.

The steam-using part of the plant is, as already noted, mounted on the trailing component of the complete locomotive. The two components are coupled together by a bronze-bushed ball joint, which can be seen to the right of figure 3. A longitudinal section through the trailing component, or condenser vehicle, is reproduced in figure 5, and half-plans, one giving a section, in figure 6. The main by which the steam is conveyed from the boiler to the turbines on the condenser vehicle is provided with an expansion joint on the boiler vehicle, and terminates in the ball joint shown at *x*, figures 5 and 6. The general characteristics of the turbine will be readily understood from the photographs reproduced in figures 8, 9 and 10. As there shown, it is of the disc and drum type. The rotor has a single-row impulse wheel at its high-pressure end, which is followed by 18 rows of reaction blading. In the turbine used for the original Ljungström locomotive, the first stage had a two-row velocity compounded wheel, and the last row of reaction blading was divided into two sections, an inner and an outer. The steam which passed through the inner section was subsequently turned round through 180° and directed back through the outer section of blading, which delivered it to the exhaust branch. This arrangement has in the present instance been replaced by a somewhat simpler construction. The designed maximum speed of the rotor is 10 500 revolutions per minute, which corresponds to an engine speed of 75 miles per hour, and (with full steam pressure) to an output of 2 000 brake horse-power. The torque is transmitted to the leading driving axle through triple-reduction gearing, housed in the casing shown in figure 5.

The turbine rotor shaft and the first pinion shaft are both hollow, and the drive is transmitted from the one to the other by a solid shaft of small diameter,

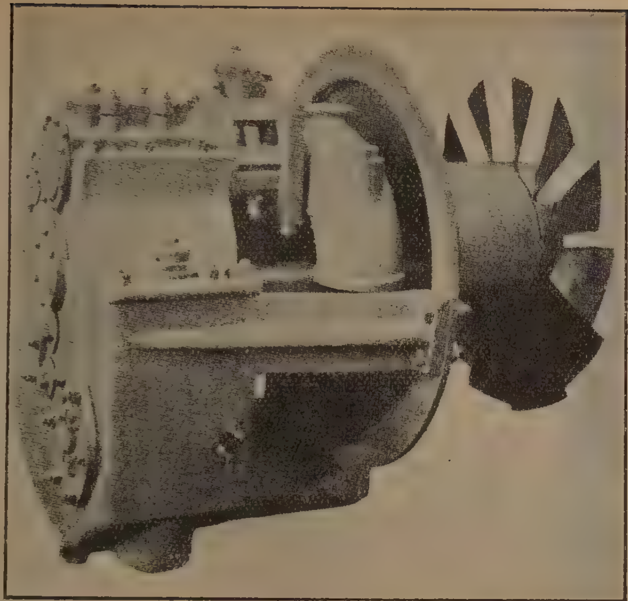


Fig. 7. — Draught turbine and fan.

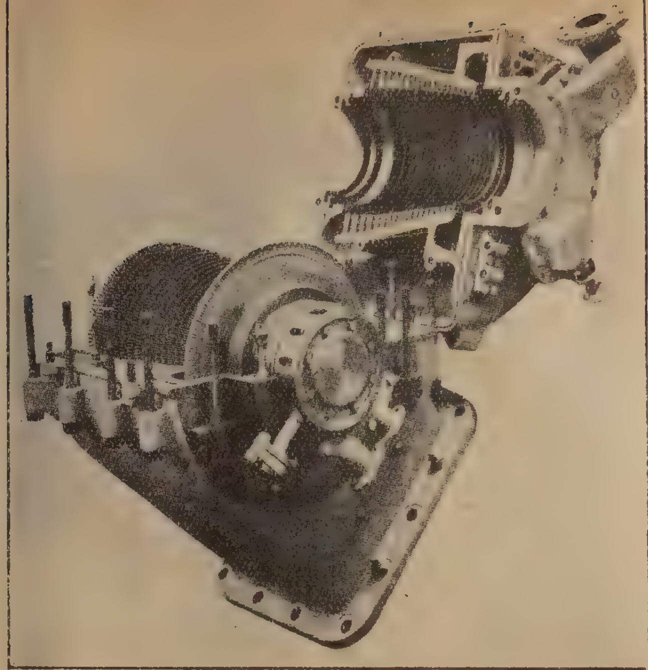
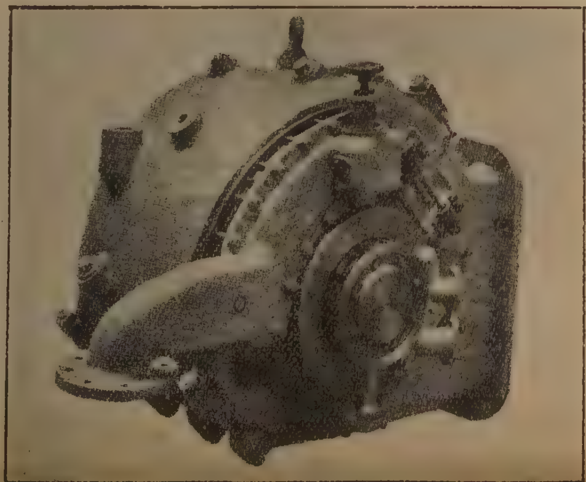


Fig. 9. — Main turbine opened up.



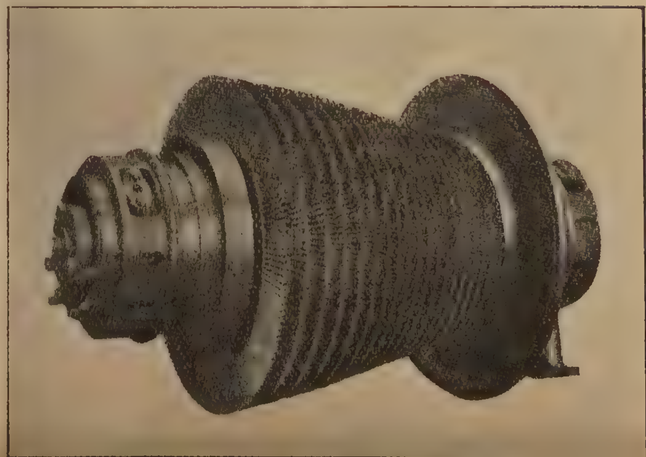


Fig. 11. — First motion shaft and pinions.

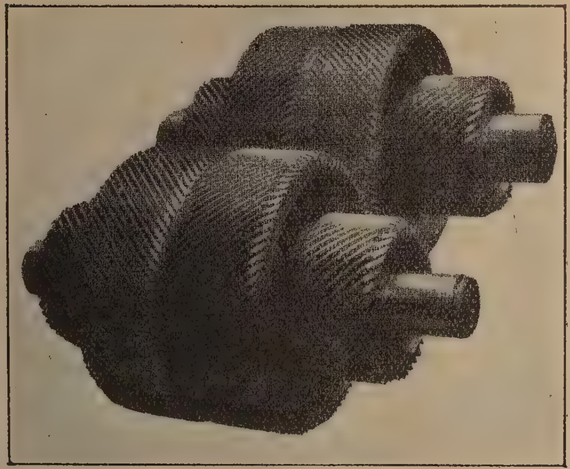


Fig. 12. — Second motion wheels and pinions.

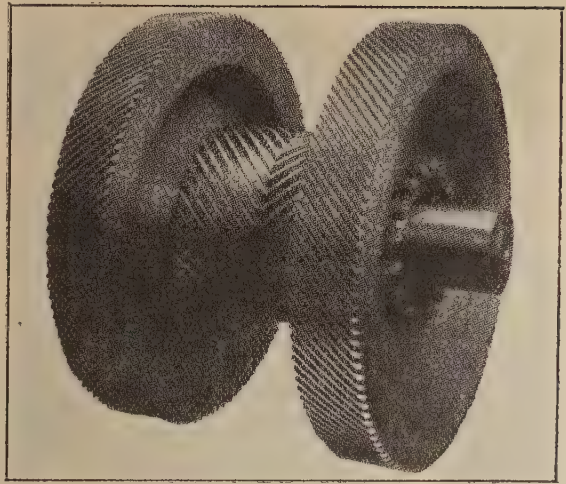


Fig. 13. — Third motion wheels and pinions.

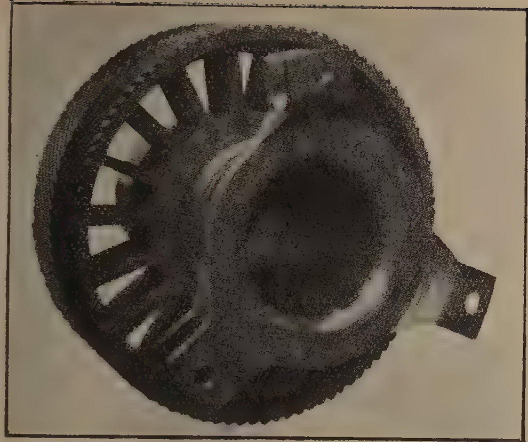


Fig. 14. — Gear wheel and quill.

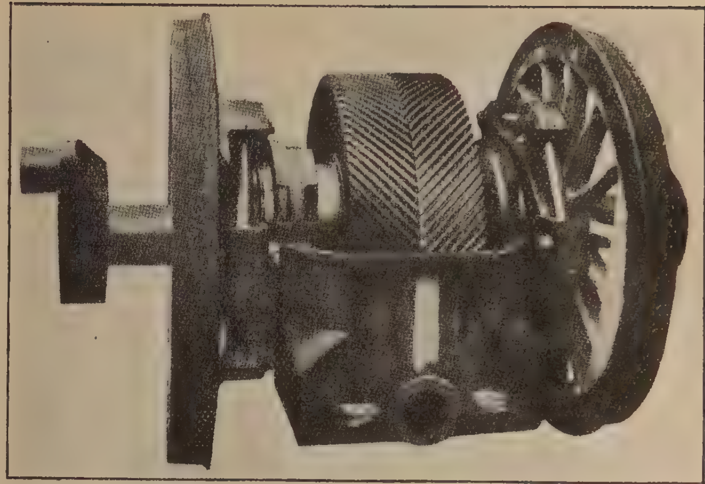


Fig. 15. — Driving axle, showing elastic drive from last gear wheel.

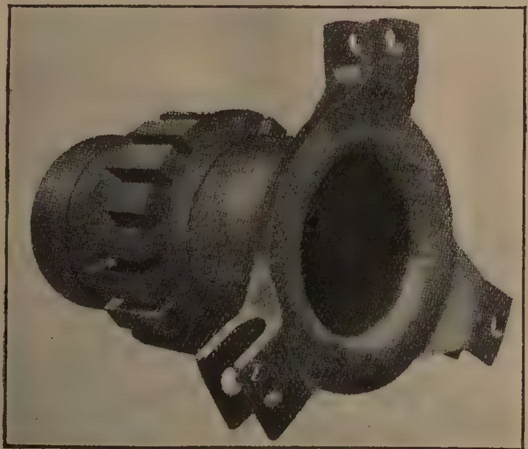


Fig. 16. — Centre for last wheel of gear.

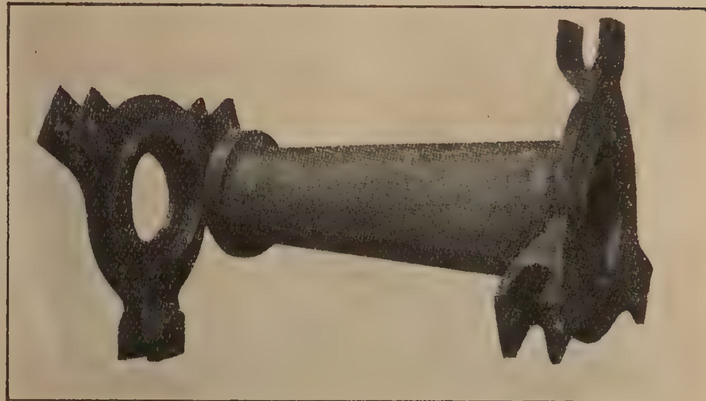


Fig. 17. — Internal quill to which driving axle is coupled.

which passes through both from end to end. At each of its extremities, it is connected by flexible discs, in the one case to the rotor shaft, and in the other to the pinion shaft. A view of this pinion shaft is reproduced in figure 11. To promote uniformity of distribution of the load over the whole length of the teeth, the metal between adjacent teeth is deeply grooved, as indicated in figure 11. Thus each tooth stands, as it were, on the top of a long thin elastic wall. This wall deflects a little under any inequality of loading, thus distributing a local excess of pressure over a considerable length of tooth. The device in question is also able to compensate for minute errors of pitch, thus compelling each tooth to take its fair share of the load.

There are two second-motion pinions, both of which are shown in figure 12. One of these pinions is idle when the locomotive is running boiler first, but comes into use when the direction of motion is reversed. The pinion teeth, as in the case of the first motion shaft, are mounted on flexible walls.

In figure 5 the gears are in the position which they occupy when the locomotive is running boiler first. As there shown, the first-motion pinions are in gear with the wheels *n* on the second-motion pinion-shaft, and the pinions on the latter are in gear with the wheels on the third-motion shaft, a photograph of these being reproduced in figure 13. The double-helical pinion on this third shaft gears with a wheel coupled flexibly to the leading driving axle of the locomotive. Photographs of this wheel are reproduced in figures 14 and 15. As there shown, the rim is carried by radial spokes, which are built up of a number of leaves of spring steel, secured at their inner ends to the hollow shaft or quill shown in figures 14 and 16. A second hollow shaft (shown separately in fig. 17) passes through that carrying the gear wheel, and this is traversed in its turn by the leading driving axle, to which it is rigidly coupled. Both the hollow shafts are provided at each end with three arms. These arms are coupled together by a linkage visible in figure 15. This transmits the drive, but allows the inner shaft and the corresponding driving axle to oscillate freely on the vehicle springs.

Referring back to figure 5 it will be seen that the reversing wheel *p* and its pinion are completely out of gear. Both *p* and *n* are mounted on eccentrics, and reversal of motion is effected by rotating these eccentrics by a weighbar and linkage operated by a screw-gear. Thus, as the eccentrics, on which *n* is mounted, are rotated, its pinions are raised out of engagement with the wheels *o*, but matters are so arranged that, during the whole of this operation, *n* remains in gear with the first motion pinion. The eccentrics on which *p* is mounted are rotated by the same motion of the reversing handle. By this rotation, *p* is brought into engagement with *n*, whilst the pinions on the same shaft as *p* are simultaneously brought into engagement with *o*. By this introduction of the idler *p* into the gear system, the direction of motion of the driving axles is reversed.

The change of gears is effected by the handle *q*, shown in figure 3, just above the main regulator handle *i*. Interlocking gear is provided, which makes it impossible to move *q* unless the regulator is completely closed and the engine brought to rest. Similarly, the regulator cannot be opened unless the handle *q* is firmly locked in one of its two extreme positions. At times the teeth may foul during the operations of gear-changing. Should this occur the motion of the handle *q* is reversed for a single turn. This automatically lowers a pawl between two of the teeth in *p*, and by means of this pawl turns *p* round by a fraction of a pitch. Hence, on resuming the original direction of the motion of *q*, the teeth fall into gear, and at the same time the pawl is lifted clear of *p* and prevent-

ed from re-engagement by a cam plate, which must be in either of one of two positions before the pawl can engage. In both running positions therefore this pawl stands quite clear of the wheel *p*.

Actual experience in running trains on the Midland line has shown that the turbine locomotive is manœuvred at least as rapidly as the ordinary engine. This has been repeatedly demonstrated in connection with the Manchester to Derby run, in which two carriages have to be picked up from a siding at Cheadle Heath. For this operation four minutes are allowed by the Company's regulations, but the Ljugström locomotive has never required so large an allowance of time.

Almost the whole remainder of the trailing section of the locomotive is occupied by the condenser and its accessories. One of the principal components of the condenser is the horizontal cylinder supported from the underframes by flexible plates, which allow of free expansion under changes of temperature. At its leading end it terminates in a steel casting to which the turbine exhaust flange is bolted. The exhaust steam is discharged directly into the cylinder, the lower half of which serves as a water tank, capable of holding about 1 350 gallons of water. The space above the water level is connected by the branches, shown in figure 5, with cast-iron headers *r* running along each side of the vehicle. Figure 6 shows also the four fans, provided for drawing the air in between the cooling tubes, by which the condensation of the steam is effected. These tubes form the sides of the condenser vehicle. They are of copper and measure 6 inches by 1/16 inch in cross section and are 5 ft. 3 in. long. In all, about 2 500 are employed to provide the cooling surface needed. Externally each tube is ribbed on both sides. The ribs make an angle with the axis of the tube, and slope different ways in adjacent tubes. The two sets of ribs, therefore, cross each other at an angle and thus, not only increase the cooling surface exposed to the air but act as spacers between the tubes. The fact that abutting ribs slope different ways serves also to promote turbulence in the air passing between the tubes, and thus increases its cooling effect. This air is drawn in by the fans shown in figure 6.

The steam exhausted from the turbine enters the condenser cylinder above water level and is met by a spray of water which falls from a perforated tray shown at *s* in figure 5. This water is raised on to the tray by the simple centrifugal pump shown at *t*, figure 5. This pump is driven by bevel gearing from the tail of the horizontal shaft which drives the four air-circulating fans. A certain amount of steam is condensed by the water spray provided as stated. The remainder passes into the upper headers and down through the flattened cooling tubes into the lower headers. The water condensed during the passage collects in the sump *u*, figure 5, from which it is raised into the condenser tank by a steam-actuated jet pump. Whilst the condensate collects in the sump *u* as just explained, the vapour and air pass on to the leading end of the lower headers. Here there are, on each side, 14 of the cooling tubes, leading up to a blanked off section of the upper header. The vapour and air are drawn up these tubes by the action of a steam jet air pump. During the passage much of the residual vapour is condensed, and is discharged into the top of the feed tank. The air also drawn off by the ejector is delivered into the same tank, whence it escapes to the atmosphere.

In service, a vacuum of 27 3/4 inches has at times been attained, but the average figure realised during a run is about 25 inches. Occasionally it may fall to 15 inches.

The four air-circulating fans have a diameter of 6 ft. 9 in. They are driven by level gearing from a horizontal shaft coupled by reduction gearing to an inde-

pendent steam turbine capable of generating 300 brake horse-power. This turbine is shown in position at *w*, figure 5. As in the case of the draught turbine, the exhaust is passed directly into the condenser.

The feed on its way to the feed pumps is passed through a two-stage feed heater located at *v*, figure 5. The steam for these heaters is derived in part from the gland leakage, in part from the vacuum brake ejectors, and in part from the exhaust of the feed pump turbine.

There are two feed pumps, one of which serves as a stand by. Both are of the two-stage centrifugal type, and are driven by a turbine designed to make 28 000 to 30 000 revolutions per minute. The high-pressure impeller is mounted directly on the turbine shaft, but the lower pressure impeller is geared down. An oil pump, of the gear type is fitted in the bottom of the casing. All the high-speed bearings have forced lubrication. The oil discharged from the bearings is collected, strained and cooled by water circulated through the cooler by the small pump shown at *z*, figure 3. The drive is derived from the draught turbine by means of chain and spur gearing.

The six trailing wheels on the boiler vehicle have steam-actuated brakes, whilst vacuum brakes are used for the six driving wheels on the condenser vehicle.

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Re-organisation of methods of carrying out repairs in the shops at the locomotive depots of the Paris Orleans Railway Company,

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(*Revue générale des chemins de fer.*)

For many years the Main Workshops of the Orleans Company have been organised in accordance with a method that Mr. Bloch, chief engineer for rolling stock and works, described in detail in an article published in the April-May and June 1925 numbers of the *Revue Générale des chemins de fer.*

The repair shops at the depots have been organised on the same principles modified to suit the peculiarities of the depots; this organisation will be described in this present article.

First of all it is desirable to give some idea of the kind and extent of repairs carried out in the shops at the depots, together with some indication as to how the work is distributed between these and the main shops. In this respect the practice of the various French systems at present differs considerably.

In principle, the repair shop at a depot carries out light and heavy running repairs to the locomotives, the engines only being sent to the main shops when requiring heavy fire-box repairs, such as the renewal of plates, repairs to boilers and serious damage resulting from an accident. In this way every engine passes through the main shops on an average once every ten years.

It can thus be seen that the role assigned to the depot shop is very important and therefore justifies the many tools provided in the shop, as well as a rational organisation of its methods.

The light and heavy repairs carried out during ordinary maintenance at the

depots are classified into two categories :

1. Daily repairs, such as re-making joints,inspection of stop valves, replacement of parts broken in service known as : « Travaux de bricole » (ordinary running repairs) ;

2. Periodical repairs which are of two kinds :

a) Piston, slide, and valve repairs called P. T. (« pistons-tiroirs »). These consist of the examination of the cylinders, slide and piston valves, and if required the replacement of any rings.

Advantage is taken of the locomotive being stopped for this purpose to take up play in the rods as well as any excessive play in the different parts of the gear; the locomotive is not lifted, however, except in cases of damage due to accident. Repairs of this kind are carried out after 35 to 40 000 km. (22 to 25 000 miles).

b) Wheels, valves and pistons repairs called R. P. T. (« roues, pistons, tiroirs »), consisting of a complete overhaul of the locomotive. The wheels are removed for turning up the tyres, all parts of the frame are examined, the play in any part is taken up. The boiler is very carefully examined and the necessary repairs made to it. These latter repairs are, however, limited to patching the firebox and wrapper, renewal of stays, palm stays and longitudinal stays of the tube plates if needed. These repairs are carried out after a mileage of 80 to 100 000 km. (50 to 62 000 miles).

It will be of interest to see how these repairs have been carried out at the depot shop, and to consider the reasons which led us to change the organisation.

Twenty years ago the repairs carried out at the sheds were relatively few : little by little they increased as the result of the larger number of locomotives, and the greater difficulty of repairing them as they became more complicated.

As the repairs increased the importance of the workshops at the depots as regards equipment and staff grew, but as it was thought no useful purpose would be served by altering their organisation, they retained the character of a shop for carrying out running repairs as required by the service. Of course it is very evident that whilst the new methods resulting from the scientific organisation of the shops increases in a high proportion the output of the men, it may also reduce the freedom of action of the depot unless attention be paid to this point. This can be a serious drawback at a depot where the repairs to be done do not occur in any regular order, and where sudden rushes of work are frequent.

Ordinary small repairs, that is to say, the maintenance work to be done daily, as booked by the drivers coming off duty, were covered by a gang under a foreman who took particulars himself of the work to be done and distributed it amongst the men. This very flexible method worked well so long as the drivers only wanted odd repairs to be done, but was not satisfactory when, after the war, handymen had to be used as drivers and did not always properly appreciate the repairs required, whilst at the same time the engines became more complicated. It was necessary at that time to control very closely the statements recorded in the repairs book and to go so far as to inspect the incoming engines by a skilled artisan who decided on the repairs needed from the information supplied by the driver.

The periodical repairs were controlled by chargemen under the orders of a foreman who took over the engines in accordance with the programme of repairs laid down and saw that the repairs were done under the best possible conditions although there was no special organisation for this purpose. As on locomotives the same defects repeat themselves, a certain unity in the method of doing the work had been obtained as a result of the direct intervention of Headquarters' inspectors. No systematic standardisa-

tion of methods had been undertaken, and owing to the lack of written orders we were not kept automatically informed as to the carrying out of the instructions given.

Therefore we could neither fix the time allowed for the same task nor make, as between the different shops, the comparisons needed in order to be able to follow the proper working of the shops.

For the following reasons : growth in importance of the shops at the depots, lowering of the professional standard of the drivers, greater complication of the engines, need of standardising the repair methods so as to introduce the most economical procedure everywhere, necessity to be able to control at all times the output of the different shops, and naturally the desire to obtain any possible increase in output of the men by improved methods of getting the work ready, it appeared to be necessary to introduce into these shops new principles of management definitely separating the preparation of the work from the execution thereof, and instituting an automatic control of the work whilst taking care to preserve in these shops the flexibility needed.

The organisation in the shops adopted with this end in view is expected therefore to cover the following points :

The laying down of definite work programmes;

The definite separation of the preparation of the work from its execution;

The issueing of written orders for each unit of production, adapting these orders to the Rowan premium system;

The standardisation of the repair methods;

The control of the carrying out of the work;

The organisation of an accounts system of the labour costs, so as to obtain prices in as great detail as possible and to automatically control the times taken.

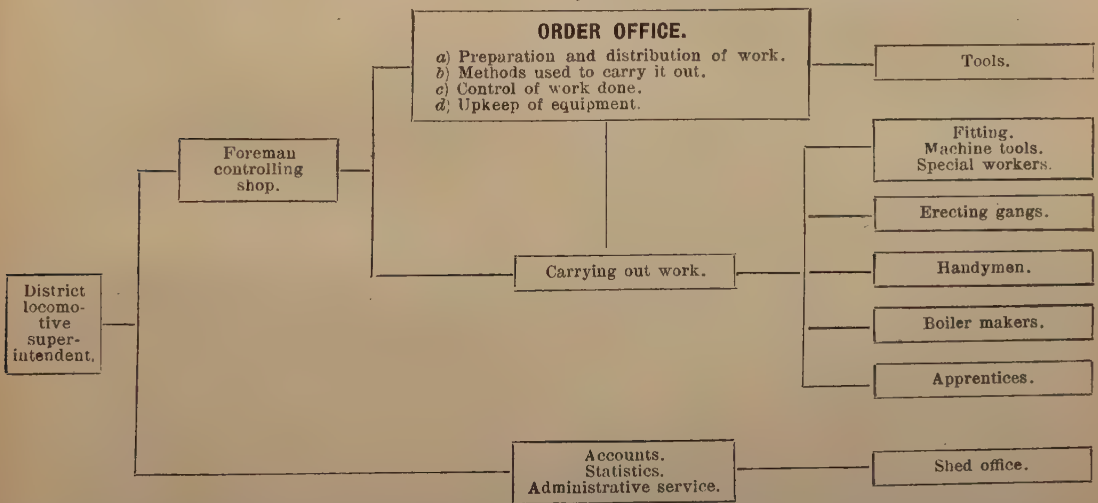
A Depot shop (1) therefore includes :

An organisation for preparing, distributing and controlling the work called « Order Office »;

An organisation for carrying out the work (various gangs of the shop);

An accounts and administrative organisation attached to the administrative office of the depot.

The organisation for dealing with repairs at a depot is as follows :



(1) Up to and including Class 2. It did not seem necessary to create any special organisation for the less important depots at which few repairs are done.

Order office.

The order office covers the following duties :

1. Preparation and distribution of the work :

a) Fixing the order in which the work is to be done; monthly programme, date engines called in, steps to be taken to enforce the stopping dates;

b) Ascertaining in detail the repairs to be done or deciding the worn parts to be replaced;

c) Seeing that supplies of the materials, parts and tools needed for the repairs are available at the proper time;

2. Technical investigation into workshop methods. Work cards. Time clocks;

3. Control of the carrying out of the work and of its quality;

4. Upkeep and development of the tool equipment.

1. Preparation of the work.

a) *Programme of repairs.* — There is no particular feature of interest in the way in which the order of carrying out locomotive repairs is decided and the monthly programme is drawn up.

Every month the district locomotive superintendent draws up the programme of the repairs to be carried out during the following month, taking into account the traffic requirements whilst endeavouring to ensure a continuous output of work from the shop.

This programme is brought to the notice of the staff concerned. The dates the engines will be stopped shewn thereon are only approximations. The actual days and times the engines will be stopped are decided four or five days in advance, and are sent in writing to the driver of the engine, to the shed assistants, and to the shop staff.

b) *Determination of the repairs to be done.* — No repair, no matter of how little importance, can be undertaken by the gangs without the order office having first of all drawn up a schedule of the work to be done.

The work schedule for any part is drawn up after the part has been examined in detail to decide very exactly the wear and tear and lays down the needed repairs after considering the design of the part to the limits prescribed for its different component parts and the most economical repair methods. The schedule oght also to lay down the materials needed in carrying out the prescribed repairs.

This schedule therefore plays a very important part in the repair, both quality and economy largely depending on it. In all cases it is prepared by the head of the order office who is an official of sufficiently high standing to take responsibility for the repairs ordered.

As soon as an engine is stopped, the head of the order office indicates the parts to be taken down : when all these pieces have been stripped down and cleaned, they are placed on a bench in a definite order (always the same) and examined by the head of the order office who says if they are to be replaced or to be repaired, and, in the latter case, decides the method of repair to be used.

In order to make the preparation of this schedule easier and to standardise the repairs, standard work schedules have been made out for each class of engine and list all the operations which can be done to the type of engine in question, and to its various parts, both when stripping down and during repairs. The schedule also gives for each operation included in the work full details with the exact wording to be written on the repair cards as well as the time allowed.

These standard schedules are of two kinds : those for stripping down, erection and repairs to the frames by the erecting gang; and those for repairs to separate parts by fitters and machinists.

The standard schedules for erection,

drawn up according to classes of engines and tenders, give in detail all repairs which can be done during a « wheels, valves and pistons » repair or when lifting a tender. These repairs are divided into two groups :

Ordinary repairs and exceptional repairs.

The ordinary repairs are those that have to be done in any case during each type of repair, whatever the general condition of the engine. Exceptional repairs, on the contrary, are only to be done when the schedule for the different parts of the frame prepared by the head of the order office shews it is desirable to do so.

The standard schedules for fitting and machine work are prepared for each class of engine or tender, and for each group of parts according to the following classification, which is the same for every class of locomotive :

Schedule.

No. 1. Pistons.
2 et 3. Valves. Piston and slide.
4. Motion.
5. Valves.
6. Injectors, feed pumps, sight feed lubricators.
7. Regulator.
8. Rods.
9 Wheels.
10. Axle boxes.
11. Horns.
12. Bogie.
13. Spring gear.
14. Reversing gear.
15. Blast pipe, blower, sandgear.
16. Cylinders.
17. Brake gear.
18. Blow down cocks, ashpan.
19. Air pumps.
20. Brake parts.
21 Drawgear, cross stays.
22. Irons, grates, drop-grate.
23. Speed recorder gear.
24. Pipe work.
25. Tender details.

These schedules give for each fitting and each part in detail the whole of the repairs that may be needed in consequence of the ascertained wear and tear. They also give the drawing numbers of the fittings, the limiting dimensions allowed, as well as technical information on the repair methods to be used and the time allowed for the different operations.

The preparation of these schedules, obviously the most complicated feature of the new organisation, represents very considerable work, seeing that we have at least thirty classes of locomotives and that each class necessitated the preparation of twenty-six standard schedules each forming a considerable volume.

These standard schedules were prepared in collaboration with all the depots between which we divided the work : they were then put in order and printed at Headquarters. The preparation of the schedules took about four years.

The role of the head of the order office when having to prepare a schedule for a repair therefore consists, after he decides a part is to be repaired, in selecting from the standard schedules the repair corresponding to the wear and tear he has discovered, and to take particulars thereof with a view to writing out the work cards. As we wished to save him all clerical work possible, he has been given a clerk to whom he dictates in turn from the schedule the number of the jobs to be carried out. This clerk writes the numbers on a special form known as the « Schedule sheet » which will provide automatically, as we shall see later on, for the job cards corresponding to the repairs proposed and to check the return of the cards as soon as the job is done. The duty of the head of the order office is thus purely technical and he has no clerical work to do.

c) *Supply of stores.* — It is very important, before stopping a locomotive for a periodical repair, to have taken all necessary steps to see the parts and mate-

rial required for the repair are available.

Particulars of these parts and materials needed as a result the wear of certain fittings can be ascertained by the reports made :

1. At the previous stopping for wheels, valves and pistons repairs (parts at the maximum limit of wear);

2. In service (various defects repaired temporarily);

3. During an inspection made a short time before the engine is stopped (defects having escaped inspection in service or abnormal wear of fittings).

As the delivery of certain parts or materials takes some little time, it is as well to have particulars about three months before stopping the engine.

With this object in view, the order office keeps for each engine a special card on which is recorded :

1. When preparing the wheels, valves and pistons schedule, all the parts which being at the maximum limit of wear will have to be renewed at the next « wheels, valves and pistons » repair;

2. In service, all defective parts both those having been temporarily repaired and those which are to be renewed at the next « wheels, valves and pistons » repair;

3. At an inspection about three months before the engine is stopped for a wheels, valves and piston repair, all parts worn or defective which had not been provided for previously.

After this last inspection the order office sends to the depot stores the list of the parts or materials required for the next repairs of the engine. The stores put on one side the parts or materials in stock and order the remainder from the area stores.

Distribution of work.

The work is distributed between the different working units by means of job cards prepared in accordance with the repair schedule.

No work can be done by any production unit of the depot repair shop (workman or gang) unless a job card, which is the authority for doing the work, has been given.

As soon as the schedule is made up, the clerk who assisted the head of the order office, makes out the job cards, which are simply a literal reproduction of the description on the standard schedule. This work is therefore simple copying and needs no technical knowledge. These cards are made out by hand at the small sheds and typewritten on rolls of paper prepared beforehand at the large depots.

These cards, as soon as written out, are sent to the chargemen of the gangs concerned who classify them in special cabinets by engine and by group of parts, and distribute them to the workmen at the proper time after inserting on each the name of the workman to whom the work is given. This indication is the only writing the chargeman has to do. Naturally the cards are not distributed to the men until the chargeman knows that the parts or materials, and also the tools required, are available.

The chargeman when giving out the cards should also take into account the date by which the job should be completed. These times are shewn on the « time schedules » drawn out beforehand in the form of a chart by the order office and sent to the chargeman, who has then nothing to do but follow them implicitly. the men until the chargeman knows that parts are received by the erectors at the exact moment they need them.

The workmen mark on the cards the times of starting and of finishing the work (except when changing from day work to premium work and inversely, when this time is marked by the chargeman) and retain them in their own possession when the work is completed. Each day the cards are collected by a special employee and sent to the depot office where after being classified by engine and kind of repair they are used, on

the one hand, to determine the number of hours spent on each repair, and on the other to check the use made of the workmen, by comparison with the time sheets of the men. Finally, the premium to be paid to each workman is calculated from them.

Thanks to the standard schedules, the amount of work done under the premium system has been greatly developed now that the methods of carrying out repairs have been laid down. In this way all erection and stripping work, all machine tool work, and all boiler work, have been priced.

Ordinary running repairs.

A special organisation was found to be needed to deal with the ordinary daily maintenance work on the locomotives known as « shed fitting ».

Each time a driver returns to the shed, he is met near the inspection pit by an inspector, a fitter, to whom he describes the repairs he thinks the engine needs. The fitter examines with him the defective parts, sees if the repairs asked for are really required after discussing them if need be with the driver who, whether agreeing or not, but under his own responsibility, enters in the special repair book the repairs which he asks should be done.

This transformation is sent to the chargeman of the shed fitters who writes out the corresponding job cards and gives them to his men, stating in particular where the engine is to be found.

When a part taken down by the shed fitters has to go to the machine shop for repairs, the shed fitter's chargeman sends it to the order office with a special card stating the work needed and the time by which he will want the part; this time is of great importance and must be observed at all costs. The order office writes out the tickets corresponding to the work and takes steps to see that the part is ready within the time given.

It would be a good thing if standard schedules, similar to those for the periodical repairs, were drawn up for the shed repairs, but this is however a task of considerable magnitude and has not yet been completed.

2. Technical investigation into the methods of carrying out the work.

As regards the investigation into the methods of carrying out the work, we find ourselves in an unusual position owing to the fact that we have to deal with a large number of depots of insufficient importance to justify the cost of a special staff for the investigation but which, as they all do work of the same kind, can receive technical instructions from a central organisation.

This « Central Investigation Office » situated at Paris, has the following principal duties :

1. To make uniform the methods of repair of the different fittings of the locomotives, so that in choosing the very best methods, standard schedules can be prepared and kept up to date.

The repairing of locomotives is an extremely complicated question, and one of a quite special technique. Putting the various parts into good order does not necessarily mean restoring them to new condition and to the dimensions of the drawings to which they were made. In the most general case, wear is allowed within limits determined by the strength of the fitting or by the work it is called on to do in any determined assembly of parts. The individual repair of the parts is therefore not the only thing to be taken into account, the question of assembly is very important.

The whole of the notes dealing with the assembly of each fitting, built up of many detail parts and the co-ordination of the fittings with one another could not be entered on the standard schedules, which are only a simple indication of the repairs that can be made to each of the

parts of the engine taken individually. It has therefore been necessary to give precise details thereof in special documents entitled « Technical notes on repairs ».

A technical note had to be prepared for each fitting which indicated in detail its condition for good working and the limiting dimensions of repair and wear to be observed. These technical notes, like the standard schedules, have been prepared by the order offices at the depots, and are then checked over and completed by the Central Investigation Office.

2. Complete preparation of certain works of general utility, such as certain notes of alterations applying to several depots, so as to obtain in all the depots a uniform method of working.

When a modification of this kind is decided on, the Central Investigation Office lays down the most economical method of carrying it out in as great detail as possible, even for certain work, pointing out the machine tools to be used and the method of setting up the work in the machines.

As soon as the best method has been decided upon, a work card giving all necessary particulars is made out and enables the depots to make the change without hesitation.

3. The preparation of technical notes is not sufficient in itself; it is necessary to see that they are acted upon. The Central Office is also responsible for this : the instructions as to their use are given, not in writing, but on the job by specially qualified employees, as it were « demonstrators ». This method of direct intervention has given excellent results : it has enabled us to have urgent alterations which had to be made at several depots within a very short period, carried out very quickly.

4. Finally, the Central Investigation Office has under its charge all standardisation questions and improvements in the equipment.

In this matter on the Orleans Railway there is complete agreement between the depots and the main area workshops. These latter, being better equipped, investigate at first hand all questions of tool equipment, quality of steel, tool forms, etc. The Central Investigation Office of the depot shops readily adopt the conclusions arrived at by the main shops and apply them to the special work of the depots.

This Central Investigation Office is therefore an essentially executive organism. It is furthermore staffed by men thoroughly trained on all repair questions, men who have spent a considerable time at a depot, and who remain in Paris for a comparatively short time, two or three years at most. These men are in constant touch with the depots at which, in fact, they spend the greater part of their time.

From this brief account, a point I should like to stress will be seen, *i. e.*, that the work has been centralised from the technical point of view, but not on the administrative side. The Central Investigation Office indicates the manner of making the repair, and this in the greatest possible detail, but it has nothing to do with either the preparatory work or the distribution of the work, which are entirely matters for the local order office.

3 and 4. Control of the work and upkeep of tool equipment.

These functions which complete the role of the order office are self-explanatory and do not call for any special remarks. This office has every facility necessary for controlling, at any moment during the repair the proper carrying out of the work.

As regards tool equipment, the order office when using it follows the lines laid down by the Central Investigation Office and is responsible, in addition, for seeing it is kept in proper repair, and renewed as necessary.

Recording and distribution of time taken.

The method of recording the time of the staff has no special features. Every morning, an employee of the administrative office, *i. e.*, independent of the executive staff of the repair shop, collects the completed cards from the workmen, and by comparing the time entered on the cards with the time on duty given by the shop, checks the way each man has been employed.

This employee also records for each workman on a special sheet the comparison of the working time with the time allotted marked on the cards with a view to arriving at the premium bonus.

The cards are then classified so as to shew at the end of the month the time taken for each periodic repair, for shed repairs to each engine, and for all other work that may have been needed.

This division, if need be, can be carried as far as desired in detail so as to determine the time taken for the repair of particular fittings, and to compare in this way different repair methods or the same repair method at two different depots.

Stores records.

The organisation we have just described has given less space to materials which have only been in question as regards anticipating needs and placing orders. The reason is that the depots, having only to carry out running repairs, use little material, and as it would involve a complicated stores system if full details had to be kept, we did not think it necessary to follow up the materials costs in the same way as the labour charges.

The Stores at the depots have a stores system which enables them to follow month by month the total expenditure of each kind of material. The depots have a unit of comparison which, other things being equal, allows the expenditure to be followed very closely : the engine-kilometre. The depot Stores prepares therefore each month by a simple calculation the cost per engine-kilometre of each kind of material taken item by item when it is a question of expensive materials, such as anti-friction metal or by groups when of less value. This information is much easier to obtain than the expenditure per individual repair and enables us to follow closey enough our *consumption* of the stores used.

Results obtained.

The improvements we have secured from this organisation are the following:

The number of hours of men employed on the repairs of locomotives shews since 1920 for the same mileage a reduction of about 40 % with less time taken for the repairs and with a better quality of work as shewn by the smaller number of accidents and the lower consumption of fuel. It is true a part of this improvement should be ascribed to the improvement in the tool equipment, and especially to the fact that the output of a large number of workmen engaged immediately after the war was low at first and has increased as they became more proficient in their work.

The comparison of the position of repairs at depots in 1921 with that before the war (1913) shews that the new organisation, together with the improvement in equipment, and whilst taking into account increases in the hourly rates of pay, has enabled us to keep the cost of running repairs per kilometre at the same figure as in 1913, although the average power of the locomotives has considerably increased, and the average tonnage of the trains has been increased by at least a third.

As this new organisation has only recently been perfected, it has not yet given its full returns : we hope in the future to get still better results.

NEW BOOKS AND PUBLICATIONS

[625 .2 (02]

NETTER (J.), Assistant Director at the French Ministry of Public Works. — **Voitures et wagons. — Matériel, freinage, éclairage, chauffage (Carriages and wagons. — Construction, brakes, lighting, heating).** — One volume in-8vo of 602 pages with 484 illustrations. — 1927, Librairie J. B. Baillière et fils, 19, rue Hautefeuille, Paris. — Price : 80 francs.

Technical literature becomes more and more markedly specialised. This is one of the consequences of the growing industrial development. Specialists look for books in which the author, whilst endeavouring to give as much useful and complete information as he can, has selected a clearly limited subject so that he can deal with it with the greatest fulness possible.

Mr. J. Netter's book belongs to this category of specialised works. It forms part of the Encyclopædia of Applied Mechanics published under the Editorship of that great authority, Mr. L. Lecornu. Mr. J. Netter is also well known by the readers of French periodicals, in which he has written many articles on railway rolling stock.

The book is divided into eleven chapters having the following titles :

Chapter I : Generalities on rolling stock;
Chapter II : Constructional details;
Chapter III : Draw gear;
Chapter IV : Body;
Chapter V : Brakes;
Chapter VI : Carriage lighting;
Chapter VII : Carriage heating;
Chapter VIII : Ferry-boats for the conveyance of rolling stock;
Chapter IX : French passenger carriages;
Chapter X : Foreign passenger carriages;
Chapter XI : Freight vehicles.

Questions under discussion at the present time with particulars and description of advanced methods of construction and of new designs will be found to have been dealt with in many places in the book.

Chapter III deals in particular with the proposed use of automatic couplers in Europe, although very great difficulties arise in connection therewith.

Chapter V gives detailed information on the application of continuous brakes to goods trains and on cab signalling.

Chapter VI describes the fittings and working features of modern train electric lighting systems.

Chapter VII deals with the heating of trains on electrically operated railways.

Chapter IX gives particulars of the most interesting features of construction and the principal dimensions of modern railway carriages, amongst which we note especially the steel carriages designed by the « Office Central d'Etudes du Matériel (O. C. E. M.) ».

Chapter X treats of American steel stock which differs from European rolling stock in its general arrangement and in the design of the frame with its main longitudinal box girder designed to take the automatic draw and buffing gear.

Chapter XI gives particulars of standard French wagons designed by the O. C. E. M.

At the end of the book the author re-

produces certain official documents of considerable value. These are :

Appendix I. The conditions laid down as regards rolling stock for international traffic including the R. I. C. regulations for carriages and the R. I. V. for wagons.

Appendix II. The French Ministerial Circular of January 1910 laying down the general regulations on the braking of trains.

Appendix III. The detailed conditions to be fulfilled by a continuous brake for goods trains as laid down at Berne on the 11 May 1909.

E. M.